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Two component heat pipe

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Department:

TWO COMPONENT HEAT PIPE

by

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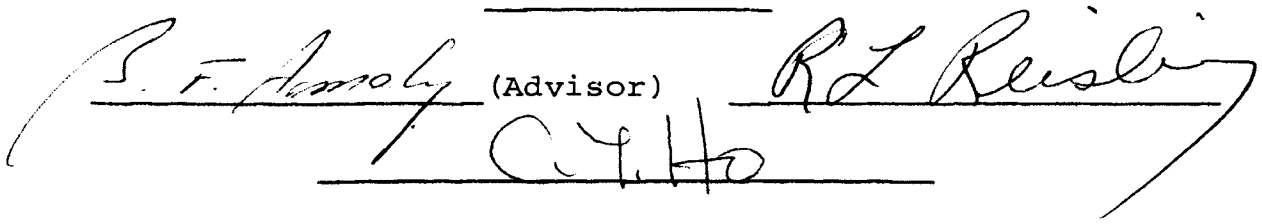

ABSTRACT

A two component heat pipe with water and ethanol as the working fluid was investigated at different angles of inclination and various ethanol concentration for two power loads. The heat pipe was 15 inches long and 0.75 inches in diameter with a 100 mesh stainless steel screen wick. The wall and the vapor temperatures were measured at different stations along the length of the heat pipe by copper-constantan thermocouples. The results were used to evaluate an existing model which includes the constraint of thermodynamic phase equilibrium of the solution on the governing equations. Based on the measured pressure and temperature distribution complete separation of the two components does not normally take place during the operation of such a heat pipe.

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The constructive criticism of Dr. R.L. Reisbig and Dr. C.Y. Ho is appreciated.

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NOMENCLATURE

Symbol	Quantity	Units
A	Cross sectional area of wick	ft^2
g	Acceleration due to gravity	ft/sec^2
h_{fg}	Latent heat of vaporization	Btu/lbm
H_e	Enthalpy of ethanol	Btu/mole
H_s	Enthalpy of Solution	Btu/mole
H_w	Enthalpy of water	Btu/mole
K	Wick permeability	ft^2
L	Total length of wicking material	ft
$\dot{m}$	Mass of flow rate	lbm/hr
M	Mass of liquid saturating the wick	lbm
ΔP_c	Capillary pumping pressure drop	lbf/ft^2
ΔP_g	Pressure drop due to gravity force	lbf/ft^2
ΔP_L	Pressure drop in liquid phase	lbf/ft^2
ΔP_v	Pressure drop in vapor phase	lbf/ft^2
P_a	Actual pressure	atm.
P_{vc}	Pure ethanol vapor pressure at T_{vc}	atm
P_{ve}	Pure water vapor pressure at T_{ve}	atm.
ΔP_m	Pressure drop due to momentum changes	lbf/ft^2
Q	Heat transfer rate	Btu/hr
r	Mean pore radius of wicking material	ft

NOMENCLATURE (continued)

Symbol	Quantity	Units
r_c	Capillary wick pore radius	ft
r_v	Radius of vapor space	ft
t	Temperature	centigrade
T_{es}	Saturated temperature of ethanol at P_a	fahrenheit
T_{vc}	Vapor temperature at condenser	fahrenheit
T_{ve}	Vapor temperature at evaporator	fahrenheit
T_{ws}	Saturated temperature of water at P_a	fahrenheit
V	Velocity	ft/sec
Y_e	Mole fraction of ethanol	dimensionless
Y_w	Mole fraction of water	dimensionless
ϕ	Inclination angle of heat pipe	degree
θ	Time	sec
μ	Viscosity of liquid	lbm/ft-hr
μ_v	Viscosity of vapor	lbm/ft-hr
ρ	Density of liquid	lbm/ft^3
ρ_v	Density of vapor	lbm/ft^3
σ	Surface tension of liquid	lbf/ft

I. INTRODUCTION

The heat pipe is an effective heat transfer device which has received considerable research efforts since 1964 when Grover and his co-workers [1] reported its successful operation. Its main advantage is that large quantities of heat can be transferred across a small temperature difference and over a long distance resulting in a very large effective thermal conductivity. It has been employed in many aspects of thermal design. It has been used for precision measurements of radiant emissivity of surfaces [2], as a thermostat for measurement of vapor pressure of liquid metal [3], and as a thermal conductor and heat flux transformer [4]. It is used in electronic circuits and in space applications requiring thermal control; space and weight being the major influencing factor for choosing the heat pipe against conventional devices. Many applications connected with high temperature and direct energy conversion are also cited in the literature [5].

In general, the heat pipe is a self contained structure made of a thin walled sealed tube. The tube acts as a container enclosing the functioning parts of the heat pipe, wick and working fluid, which lends it structural rigidity. The inner wall of the tube is lined with a thin layer of porous material, called a wick, which is usually saturated

with the working fluid. The heat pipe length is usually divided into an evaporator section, adiabatic section and a condenser section. The adiabatic section when present, separates the evaporator from the condenser section. Thermal energy from an external source is introduced through the evaporator wall to the saturated working fluid causing it to evaporate. The vapor moves, under the influence of small pressure gradient, to the condenser section through the core of the heat pipe. The vapor condenses at the condenser section and the thermal energy released during this process is transferred to an external sink through the tube wall. The condensate, from the condenser, returns to the evaporator section through the capillary wick utilizing the surface tension forces of the liquid. This cycle, of evaporization and condensation, is repeated continuously during the steady state operation of the heat pipe.

The above described heat pipe cycle where thermal energy is transferred from a source to a sink by continuous mass cycling and phase change of a suitable working fluid is common in many engineering systems. The coffee percolator, gravity induced mass cycling, and distillation column pump induced mass cycling, are a sample of these systems. The unique feature of the heat pipe is the fact that the mass cycling is achieved by the surface tension forces of the fluid and thus eliminating the need for a pump or gravity field. The operating pressure in a heat pipe is

dependent on the vapor temperature, where it is an independent parameter (usually constant at one atmosphere) for the other type of similar systems. The high energy transfer associated with the phase changes of the cycling fluid makes this device suitable for application where high rates of energy are required to be transferred across a small temperature difference. The temperature range covered by a given heat pipe depend on the operating fluid and can cover the range from the triple point to the critical point. Heat pipes have been shown to operate successfully at very high temperatures with liquid metals [6] and at cryogenic temperatures with liquid nitrogen [7-8].

In principle by providing a sufficient heat transfer area for both the condenser and evaporator sections the heat pipe can transport a wide range of heat transfer rates. A heat transfer rate is eventually reached at which the heat pipe starts to suffer from a rise in temperature at the evaporator section and as a result the heat pipe ceases to be practically operable. The maximum capability is usually characterized by this limiting heat transfer rate. This limitation could be due to many factors such as, the limiting capacity of the wick capillary pumping head, vapor lock in the evaporator section, shocked vapor flow in the core and the presence of non-condensable foreign gas. The last cause may be due to chemical reaction between the fluid and the remaining structure resulting in gas generation or

due to accidentally or purposely introducing foreign gas during the filling process. In order to detect and control the behavior of such a device when the foreign gas is present it is essential to examine the operating characteristics of multi-component (more than one working fluid) heat pipe.

The present study examines the operating characteristics of a two component heat pipe. Water and ethanol at various concentration was used as a working fluid. The operating principles of such a heat pipe is the same as those governing the operation of a single component heat pipe with the exception that two components are actively participating in the heat pipe processes, such as evaporation and condensation. This modification will add the constraint of phase equilibrium to the governing parameters. The operating temperature range (usually characterized for a single component by the range between the critical and the triple point) for two component heat pipe will be in the overlapping region of the individual component. For example, a water-ethanol heat pipe could have both components participating as heat pipe fluids in the temperature range from $930^{o}R$ to $492^{o}R$, in which the two operating temperature ranges (for water 1165^{o} - $492^{o}R$, and for ethanol 930^{o} - $113^{o}R$) overlap. Outside the specified temperature range the heat pipe cannot be regarded as a functioning two-component heat pipe. At steady state partial separation of

the two components will take place leaving a rich water solution, less volatile, at the evaporator section and rich ethanol solution, more volatile, at the condenser section. Due to differences in concentrations an appreciable temperature difference will exist between the two sections. In effect two zones of approximately constant but different temperatures could exist along its length. The heat pipe will operate in such a manner as to maintain constant total pressure along its length and the temperature will vary in accordance with the thermodynamic phase equilibrium of the binary mixture.

The object of this experimental study is to evaluate the above model and to examine the condition where complete separation can exist by considering different concentrations of water ethanol solutions.

II. REVIEW OF LITERATURE

In recent years a large number of articles, well over a hundred, on heat pipes have appeared in the literature. The major portion deal with one component heat pipes operating at high or moderate temperatures. Only a few articles dealt with low temperature or two component heat pipes. A detailed review of the literature can be found in an article by Winter and Barsch [5]. The following is only a few of the major references describing the development in this area.

The first concept of a heat pipe was proposed by Gaugler [9] in 1944 and re-evaluated by Trefethen [10] in 1962. Unfortunately, neither one pursued its complete development and the verification of its principles. In 1964 Grover [1] and Cotter [11] experimentally and theoretically formulated and verified the principle of such a device. The work of Cotter still is the basis for most theoretical work in this area. The heat pipe was then defined as a closed structure containing some working fluid which transport thermal energy from one part of the structure to another part by means of vaporization of the liquid, transport and condensation of the vapor, and subsequent return of the condensate from the condenser by capillary action to the evaporator.

The basic theory for heat pipe design which was formulated and verified by Cotter balances the pressure drop in the system as

$$\Delta P_c \geq \Delta P_v + \Delta P_L + \Delta P_g + \Delta P_m \qquad (1)$$

where ΔP_c is the capillary pumping pressure head, ΔP_v is the vapor pressure drop, ΔP_L liquid pressure drop, ΔP_m momentum pressure drop and ΔP_g is the pressure drop due to gravity. The heat pipe will operate successfully if the capillary pumping pressure is sufficient to overcome the pressure losses caused by vapor transport, viscous drag, gravity field and momentum changes. Each of these pressure drops could be predicted using the following relations. For example, the capillary pumping head is given by

$$\Delta P_c = \frac{2\sigma}{r_c} \qquad (2)$$

The viscous pressure drop based on laminar flow through porous media is given by

$$\Delta P_L = \frac{\mu L \dot{m}'}{\rho K A} \qquad (3)$$

The vapor pressure drop based on laminar flow through a tube is given by

$$\Delta P_v = \frac{4\mu_v Q L}{\pi \rho_v r_v^4 h_{fg}} \qquad (4)$$

This value has been shown to be small in relation to other terms in the pressure balance equation and usually is neglected. The pressure drop due to gravity field is given by

$$\Delta P_g = \pm \rho g L \cos\phi \tag{5}$$

The (+) sign is used when the evaporator is below the condenser section and (-) sign when the condenser is below the evaporator section. The pressure drop due to momentum changes is given by

$$\Delta P_m = \frac{M}{A} \frac{dv}{d\theta} \tag{6}$$

This term has been shown to be small relative to the other pressure drop terms and usually is neglected.

By substituting the significant pressure drop terms ΔP_L, ΔP_c, ΔP_g, in equation (1) an expression for the maximum flow rate in the wick can be obtained as

$$\dot{m}' = \frac{\rho K A}{\mu L} \left(\frac{2\sigma}{r} \pm \rho g L \cos\phi\right) \tag{7}$$

By assuming that the total energy transfer across the length of the heat pipe is only due to the mass cycling, through the evaporation and condensation processes, the maximum heat transfer can be predicted by

$$Q_{max} = \dot{m}' h_{fg} \tag{8}$$

The above theoretical development is limited by the fact that some of the properties appearing in the equations could not be determined and are not readily available. For example, the permeability of the wick and the capillary wick radius have to be evaluated experimentally for the geometry under consideration. Also the above formulation assumes uniform temperature along the length of the heat pipe, a case which is violated in most investigations. These limitations indicate that for accurate prediction of the heat pipe performance, an experimental evaluation is needed.

The capability of the heat pipe could be limited or controlled by various means which have been used effectively in thermal design applications. For example, the flow of the vapor or the liquid could be restricted [1] resulting in reducing the heat flow capability and thus using the heat pipe as variable thermal conductivity link. Another technique is the intentional introduction of noncondensable gas into the vapor space. The gas tends to collect in the condenser where it forms a relatively stagnant gas zone which effectively reduces the condensation area and rate. Katzoff [12], Wyatt [13], and Anand [14] have suggested the use of a thermostatically controlled valve to supply or withdraw the noncondensable gas. Such an arrangement would allow the effective condenser area to be varied independently of the operating pressure prevailing in the pipe and could

be used to control its operation.

The heat pipe may be applied to a number of thermal transport problems. Katzoff together with several other investigators [12] recommended the use of long heat pipes, wrapped around the circumference of spacecraft, to accomplish the necessary equalization of the temperature distribution. Anand [15] reported the successful employment of two Freon-11 heat pipes. Deverall [2] utilized the heat pipe to measure the hemispherical emissivity of variously prepared surface. Schretzmann [16] employed the isothermal surface of a heat pipe as a source in a study of the effect of electromagnetic fields on the evaporation of metals. Bohdansky and Schins [3] used a heat pipe to determine pressure-temperature relations of vapor metal at high temperatures and pressures. Feldman and Whitney [17] suggested the construction of an isothermal flat for the installation of electronic components. Many investigators [18] have considered its application to energy dissipating systems and have cited numerous advantages including greater heat transfer per unit weight. Deverall [2] reported that a silver heat pipe was 520 times more effective than an equivalent solid tantalum rod.

At cryogenic temperatures the heat pipe has been investigated experimentally by Haskin [7] and Dudheker [8] and theoretically by Chi [19]. The experimental investigation dealt mainly with nitrogen heat pipe and it indicates that

such a heat pipe could be used effectively as a thermal link with a conductivity better than copper.

The two component heat pipe was first evaluated by Cotter [11] and Katzoff [12] for the purpose of thermal control. Neither one, however, performed experimental evaluation of such a device. Both assumed that the two components will separate completely and the heat pipe will perform as two individual heat pipes. Tien [20] was the first to realize and confirm that the above model was wrong, and proposed a new one based on thermodynamic phase equilibrium where complete separation will rarely take place. Water-ethanol heat pipe was evaluated by Tien [22] and Feldman [23] examined experimentally water-methanol heat pipe.

III. THERMODYNAMIC PHASE EQUILIBRIUM

The composition of the phase in equilibrium when two or more components are in solution is of paramount importance in a great number of physical processes. The most common of these processes is in the fractional distillation where the more volatile component in the solution is separated from the bulk by successive condensation and vaporization. The two component heat pipe which is under consideration here, uses a process similar to the fractional distillation. The main difference between the two is the role of the pressure which is maintained constant, in the fractional distillation and is usually atmospheric. In a heat pipe the pressure is a variable, corresponding to the equilibrium pressure, and is dependent on the temperature and the composition of the vapor phase. This difference makes the heat pipe process a more difficult one to analyze and to predict. The two component heat pipe involves a continuous process of evaporation and condensation, resulting in a liquid and vapor phase in equilibrium. The properties of the phase will dictate the operating characteristics of such a heat pipe. Unfortunately, properties such as enthalpy and latent heat are not usually tabulated for a two component solution and the calculations should be performed for the specific case under consideration. This section deals with the method of obtaining the magnitude

of these properties using thermodynamic relations and some existing experimental phase equilibrium data.

The treatment of equilibrium between gas and liquid in a two component solution could be quite simple if each phase exhibited the properties of an ideal solution. Under these ideal conditions the composition of the phase in equilibrium could be computed by using the fugacities of the pure component. The behavior of most solutions, however, deviate markedly from that of ideal solution and as such experimental results are needed to obtain accurate phase equilibrium composition. For water-ethanol solution this information can be found in reference [23] and is presented graphically for different pressure in Fig. 1 and Fig. 2. The data in Fig. 2 was interpolated from Fig. 1 for the pressures used in the experiment. For any given temperature and a fixed pressure the composition in the liquid and the vapor phase can be read directly from the figures.

Using thermodynamic relations the enthalpy of the solution can be expressed as

$$H_s = Y_e H_e + Y_w H_w \tag{9}$$

where H stands for the enthalpy and Y stands for the mole fraction. The subscripts s, e and w stand for the solution, the ethanol and the water, respectively. The latent heat can be calculated by subtracting the enthalpy of the gas from that of the liquid at the same temperature and pressure resulting in the following relation.

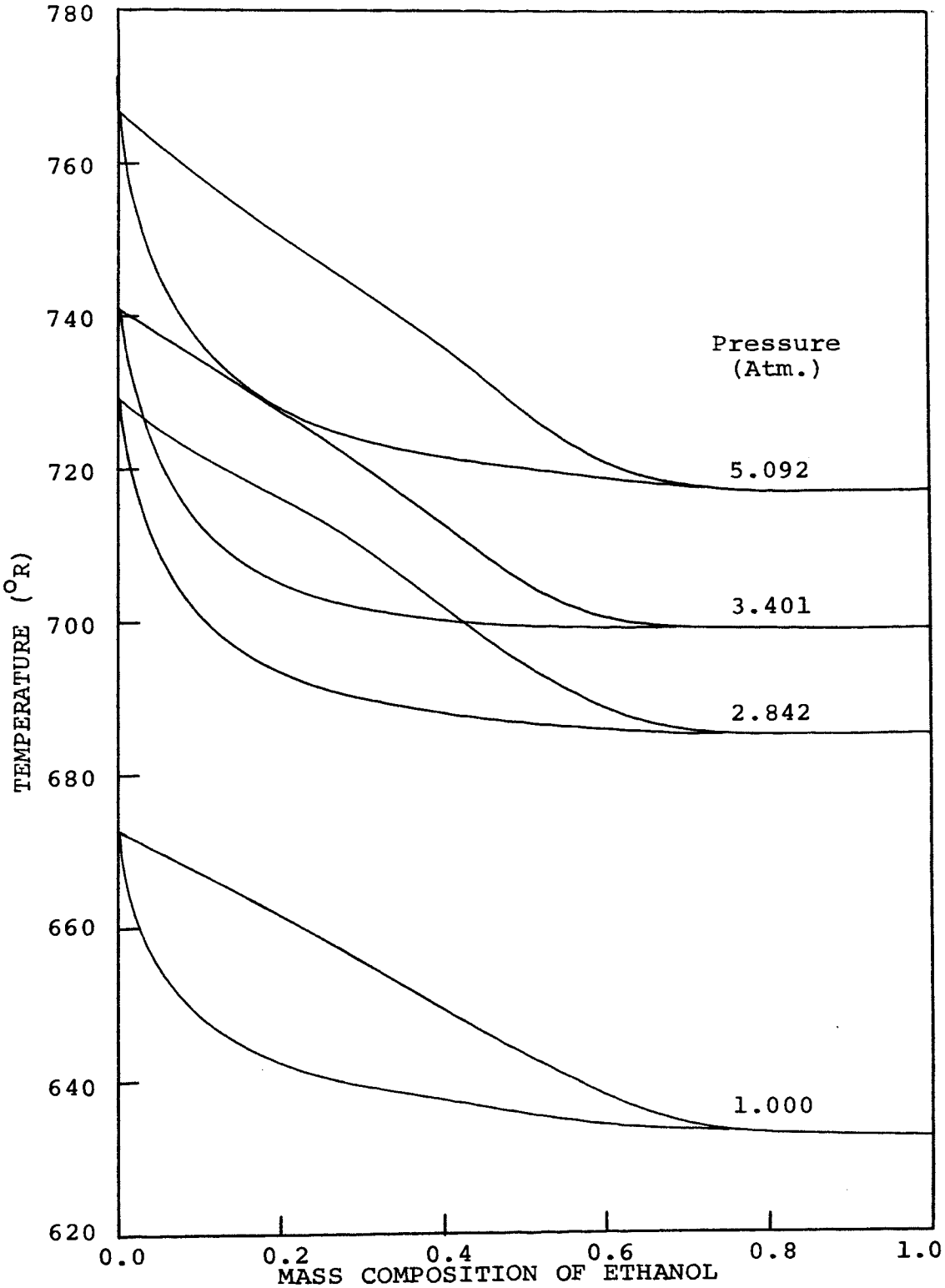

Fig. 1 Phase Equilibrium Diagram for Water-Ethanol

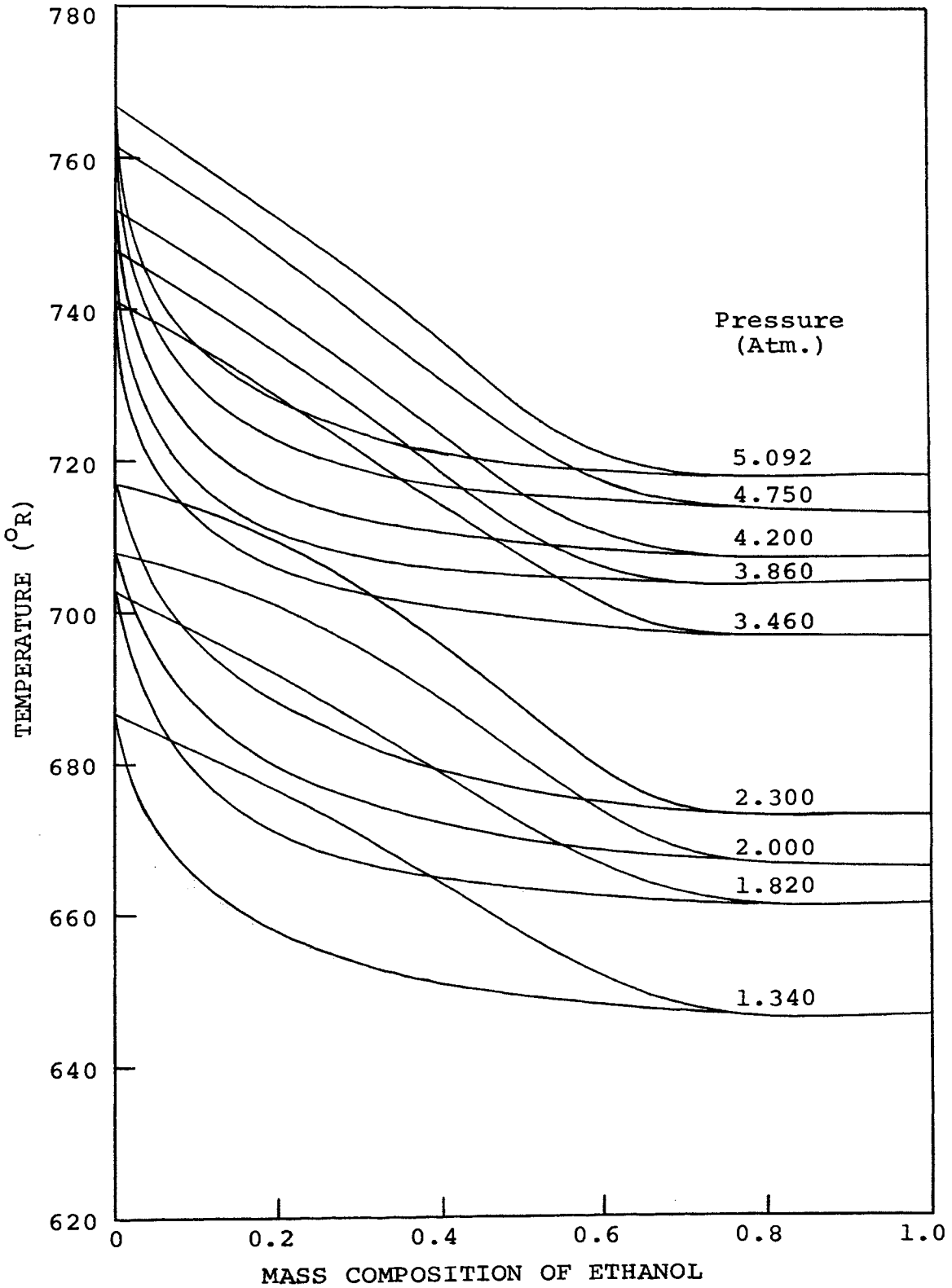

Fig. 2 Phase Equilibrium Diagram for Water-Ethanol by Interpolation

$$H_{fg} = H_g - H_f = Y_{eg}(H_{eg} - \frac{Y_{ef}}{Y_{eg}} H_{ef}) + Y_{wg}(H_{wg} - \frac{Y_{wf}}{Y_{wg}} H_{wf}) \quad (10)$$

where the subscripts g and f refer to the gas and the liquid phase respectively. For the calculation of latent heat from the above equation, the properties of both water and ethanol and the composition of vapor and liquid phase should be known. The composition in the gaseous phase Y_{eg} and Y_{wg} and the liquid phase Y_{ef} and Y_{wf} can be taken from Fig. 1 for a specified pressure and equilibrium temperature. The properties of water can be taken directly from reference [24] and the ethanol properties will be calculated using the following outline procedure. (Ethanol properties are not readily available in tabulated form in the literature).

The saturated pressure for ethanol can be calculated using Antonine [25] formula which is given by

$$\log_{10} P = A - \frac{B}{t+C} \quad (11)$$

where for ethanol A = 8.04494, B = 1554.3, C = 222.65, t is in degrees centigrade and P is in millimeter of mercury. Fig. 3 shows a plot of the saturation pressure versus temperature resulting from the above expression. The enthalpy of the saturated liquid ethanol at a given pressure can be calculated using parabolic relation for the specific heat [25] which is valid over the experimental temperature range and is given by

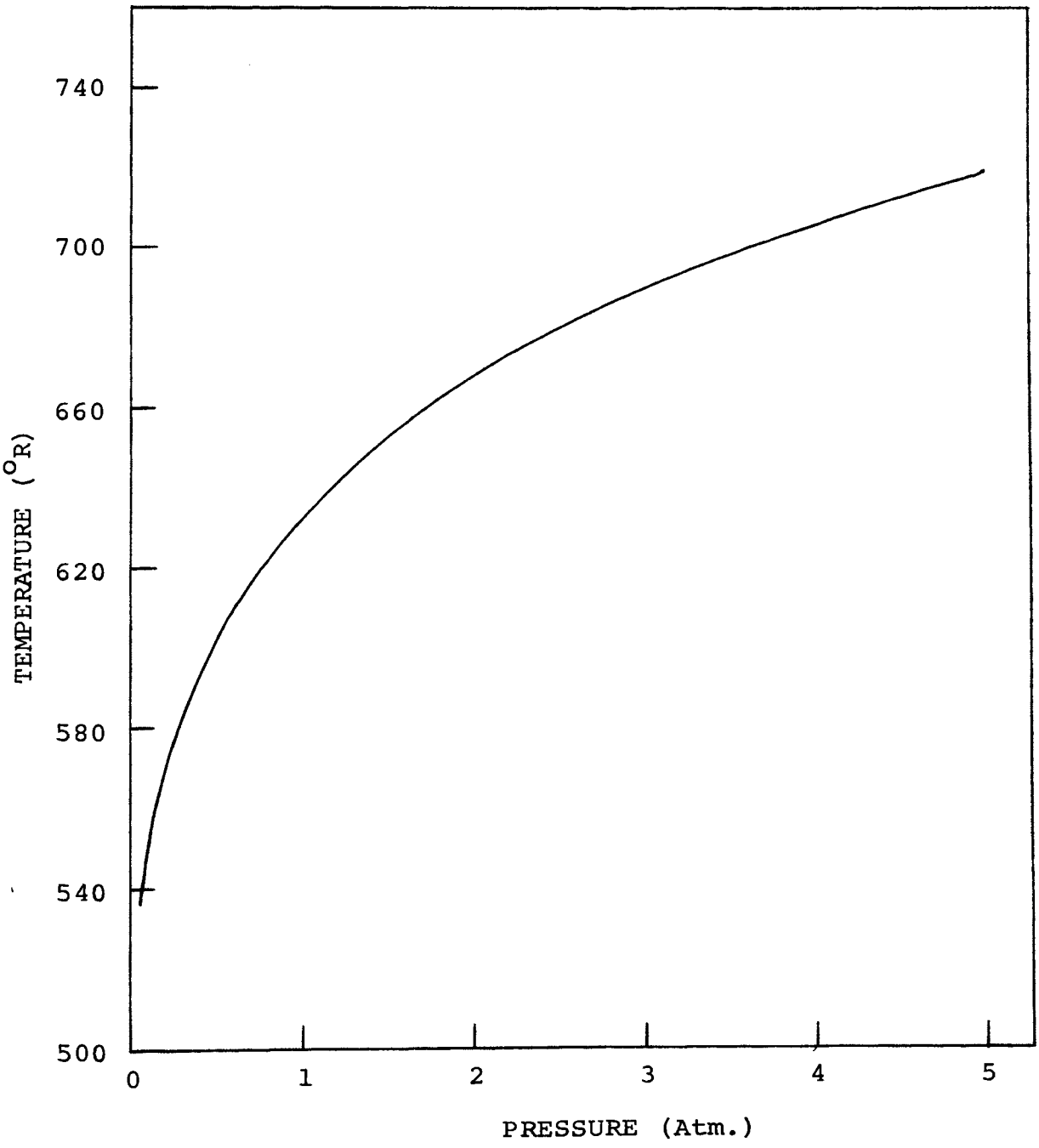

Fig. 3 Saturated Pressure of Ethanol

$$C_p = -10.83 + 0.118T + 0.000025\ T^2 \qquad (12)$$

where T is in degrees Kelvin and C_p is in (Cal/mol-^oC). The enthalpy of the liquid at any given temperature can then be evaluated from

$$H_{ef} = \int_{273}^{T} C_p dT = -10.83(T-273) + 0.059[T^2 - (273)^2] + 0.0000083[T^3 - (273)^3] \qquad (13)$$

where as a reference, the enthalpy at temperature 273^{o}K was taken as zero $H_{273} = 0$.

The enthalpy of the pure enthanol saturated vapor can be calculated by adding the latent heat to the liquid enthalpy resulting in

$$H_{eg} = H_{ef} + H_{efg}$$

where H_{efg} is the latent heat calculated from the modified Clausius Clapeyron equation [25] which is given by

$$H_{efg} = \frac{RT^2}{P} \left(\frac{dP}{dt}\right)_s \left[1 - \frac{T_c^3\ P}{T^3 P_c}\right]^{1/2} \qquad (15)$$

where R is the universal gas constant (1.986 Cal/gm-mol-^{o}K), P saturated pressure (atm.), T saturated temperature (^{o}K), T_c critical temperature (^{o}K), P_c critical pressure (atm.). Using Antonine formula, equation (11), the derivative of

the saturated pressure with respect to temperature appearing in the above equation can be evaluated resulting in

$$\left(\frac{dP}{dt}\right)_s = \frac{2.303\ BP}{(T + C - 273.15)^2} \qquad (16)$$

and by substituting in the governing equation and using appropriate constants for ethanol, it is possible to deduce the following relation for the latent heat

$$H_{efg} = \frac{4.5757\ T^2B}{(T + C - 273.15)^2}\ \left[1 - \frac{(516)^3P}{T^3 \times 63}\right]^{1/2} \qquad (17)$$

Appendix A contains the ethanol properties versus temperature calculated using the above procedure.

The properties of ethanol and the properties of water were used to calculate the latent heat of the solution for a fixed pressure while varying the concentration by using equation (10). The results of these calculations for different equilibrium pressure are presented graphically in Fig. 4 and Fig. 5. It is interesting to note that the latent heat has a peak for a given saturated pressure and that the magnitude of the latent heat decreases with increasing pressure and with increasing ethanol concentration beyond the peak value.

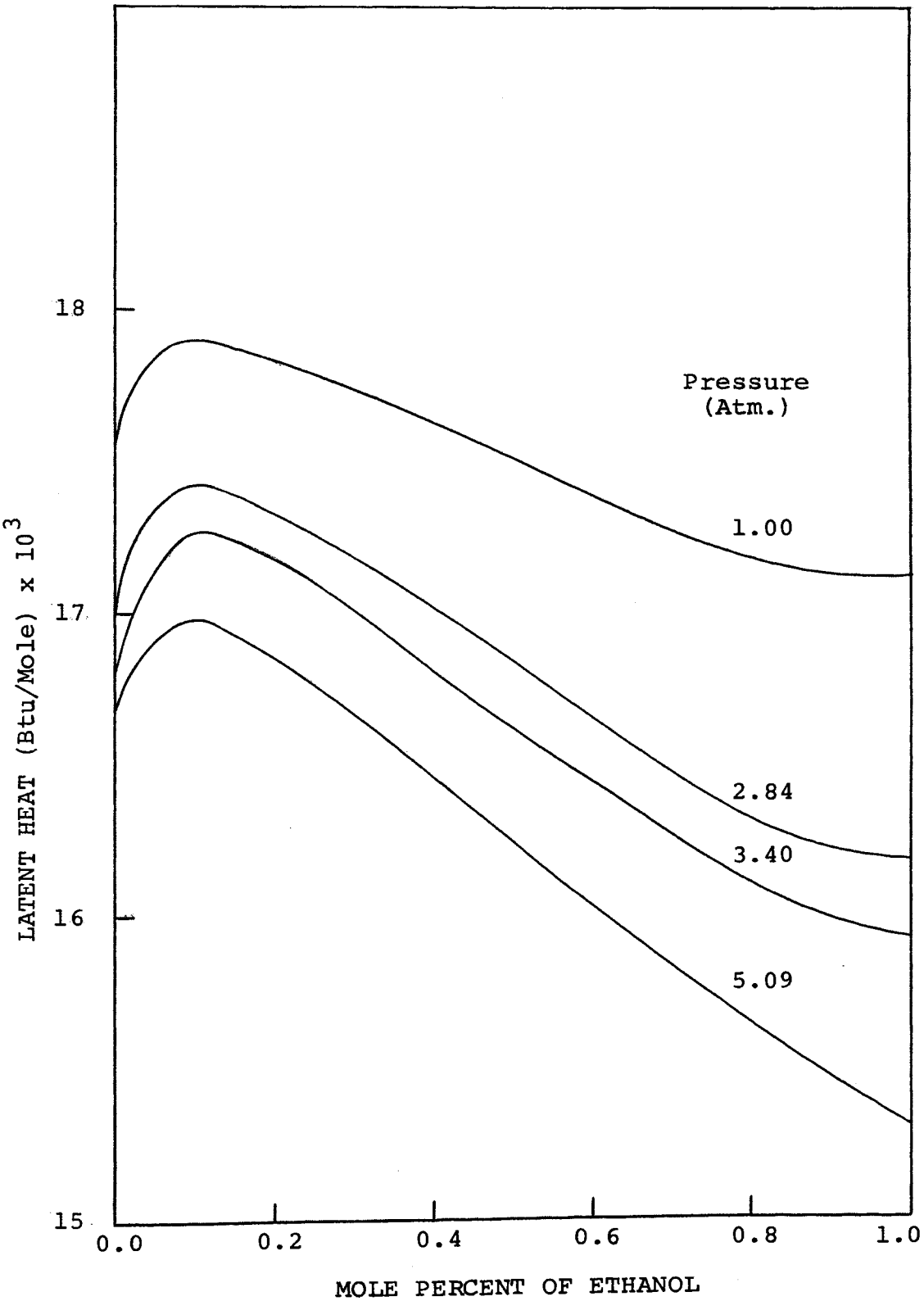

Fig. 4 Latent Heat for Water-Ethanol Solution

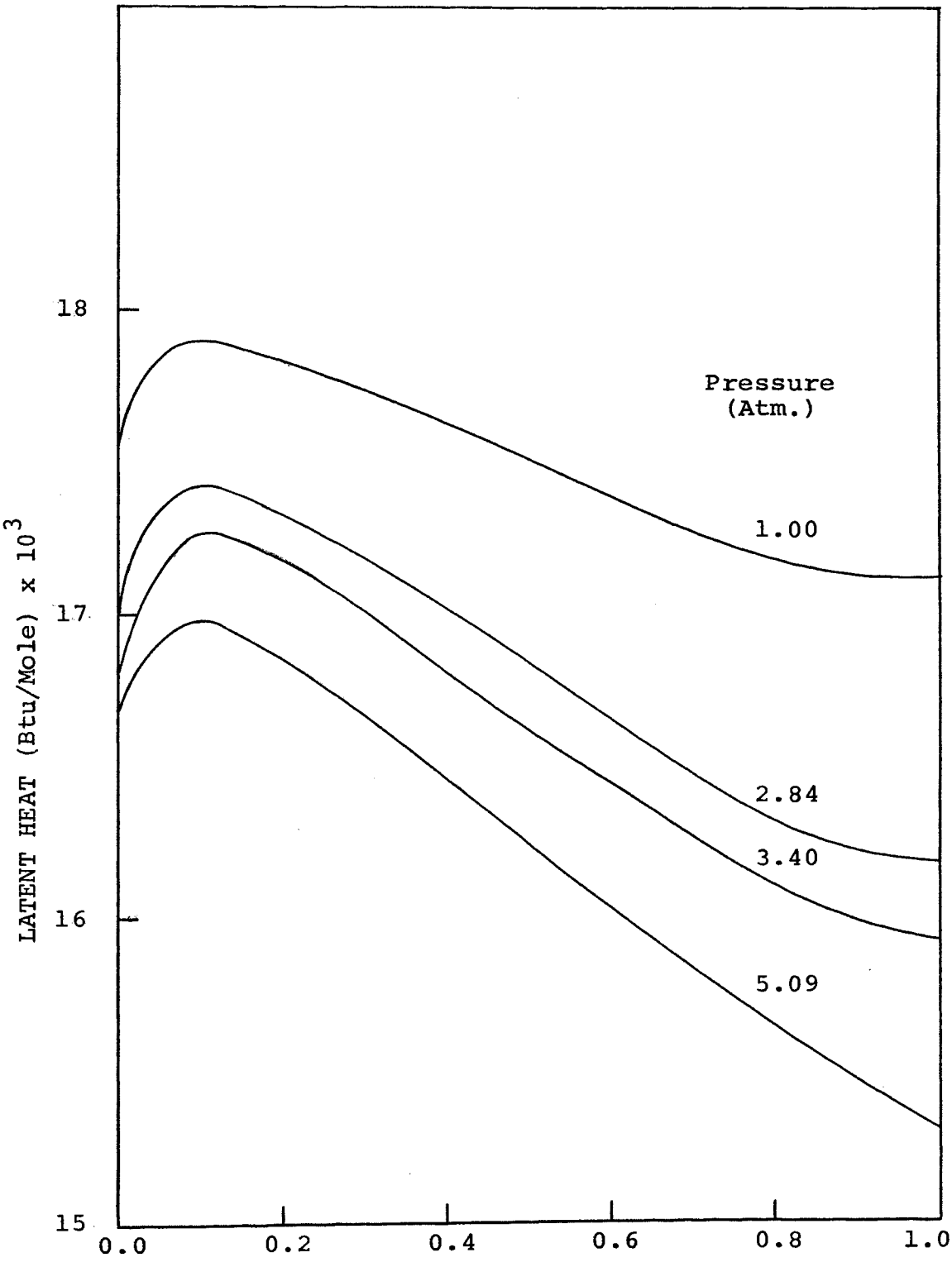

Fig. 4 Latent Heat for Water-Ethanol Solution

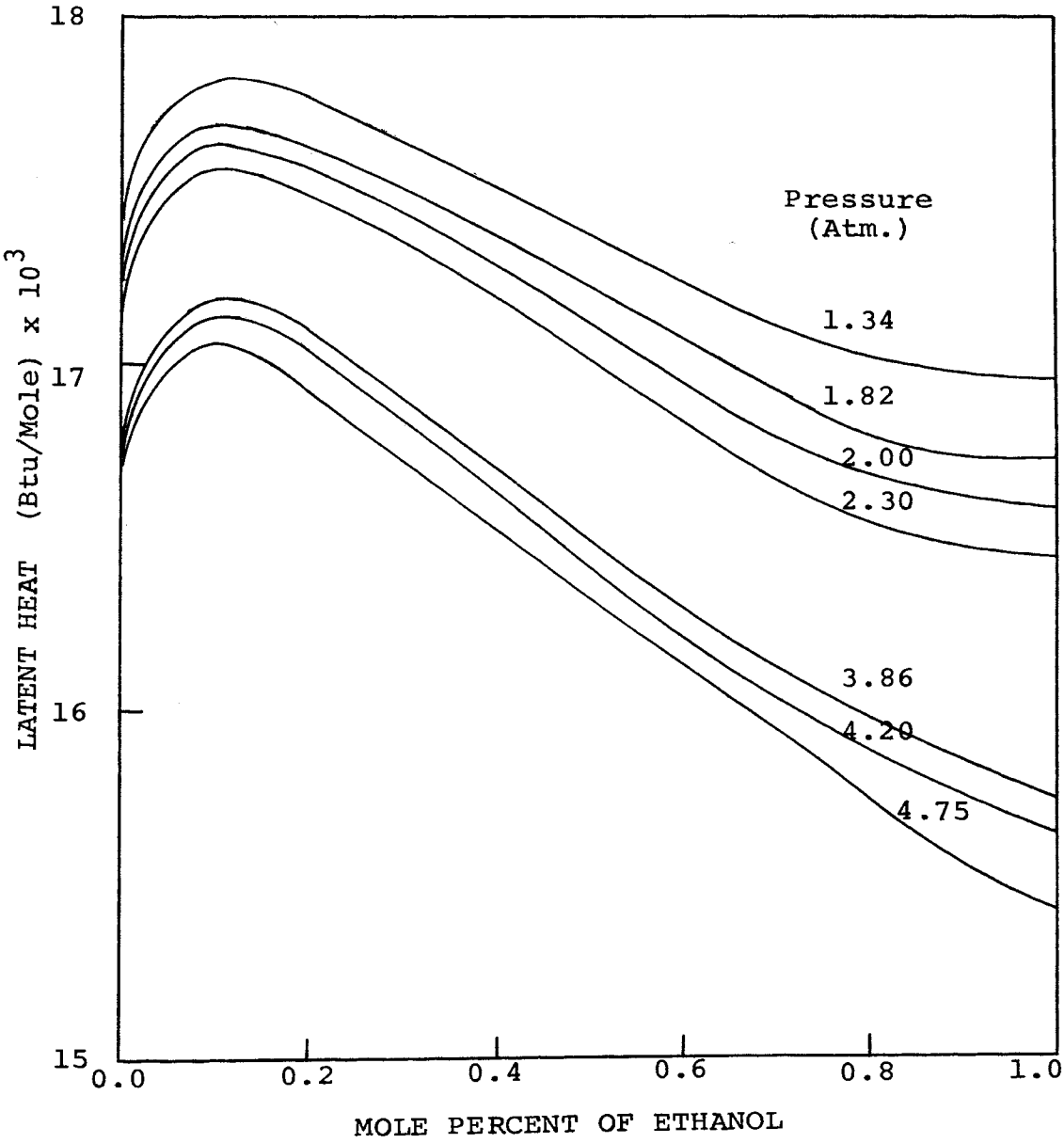

Fig. 5 Latent Heat for Water-Ethanol Solution by Interpolation

IV. DESCRIPTION OF APPARATUS AND EXPERIMENTAL PROCEDURE

The heat pipe was constructed from a 22 guage stainless steel, type 304, tubing with an outside diameter of 0.75 inches and was 15 inches long. Two stainless steel flanges were soldered at the two ends to form a closed containing tube. A 100 mesh stainless steel screen, three layers thick, was used as a wick and was held tightly to the inner wall of the containing tube by a helical spring.

The evaporator section of the pipe was 4 inches long at one end of the tube. A copper sleeve, 0.25 inches wall thickness, was fitted tightly over that section to maintain temperature uniformity along its length. Three holes, 0.5 inches in diameter, were drilled into the sleeve wall, and were distributed uniformly along its length, to allow thermocouples connections to the heat pipe surface. Electrical energy can be dissipated through a nichrome wire resistance heater which was wrapped uniformly and cemented to the outer surface of the sleeve by General Electric high temperature cement. The remaining 11 inches of the heat pipe acted as a condenser section.

The axial temperature distribution along the outside surface of the heat pipe was measured by fifteen copper-constantan thermocouples located along the heat pipe length as shown in Fig. 6. The axial vapor temperature distribution

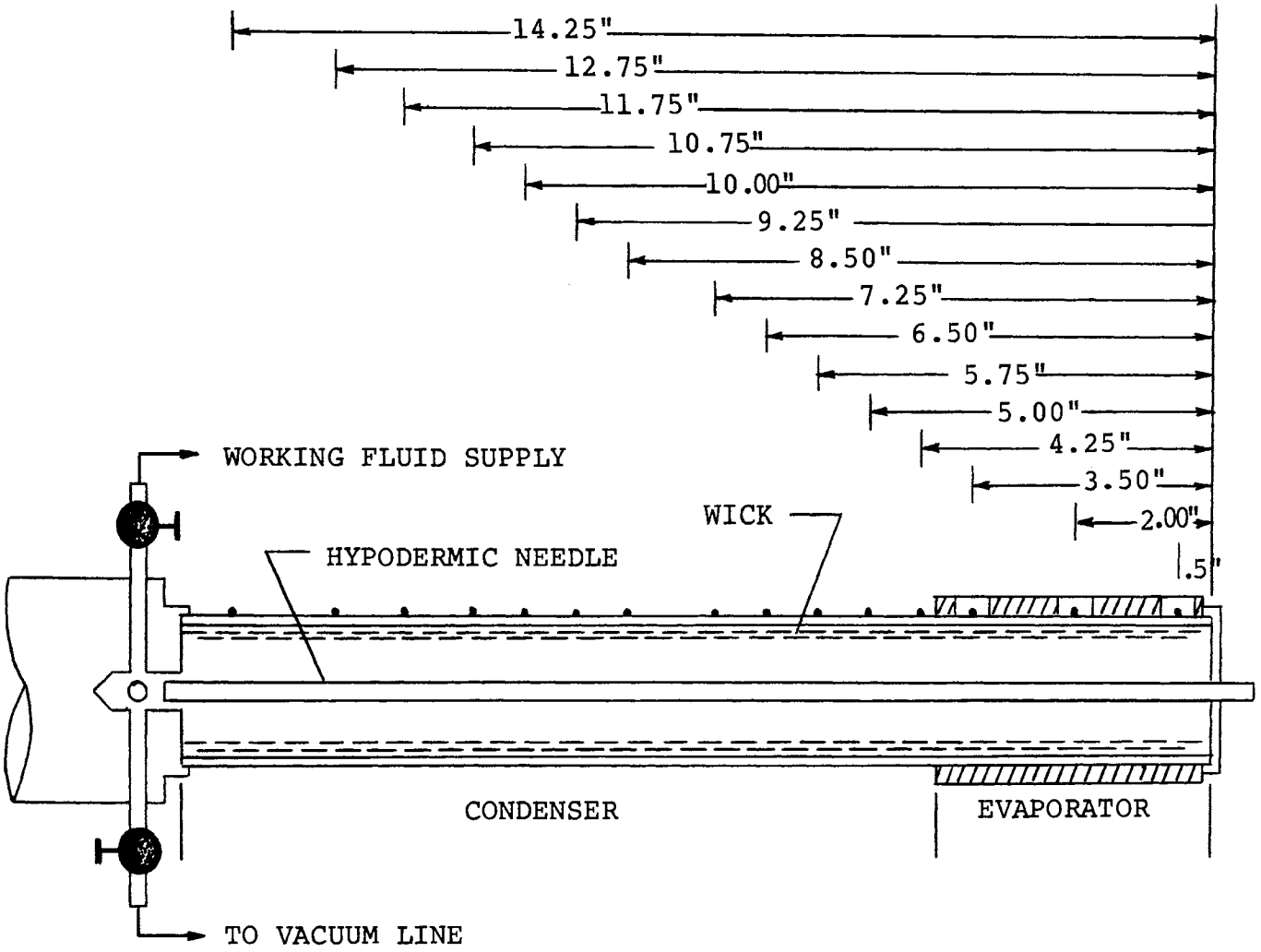

Fig. 6 External Thermocouple Location

inside the heat pipe was measured by an additional eight copper-constantan thermocouples. These thermocouples were placed inside a 0.125 inches diameter thin wall stainless steel tubing (a long hypodermic needle), which was in the center of the heat pipe and extended over its entire length as shown in Fig. 7. All thermocouples were 5 mils in diameter and were soldered at the appropriate section along the length of the heat pipe.

The heat pipe was constructed in such a manner as to permit its support in a device which can be rotated and fixed at any angle of inclination. This assembly was used to investigate the affect of inclination angle on the operating characteristics of the heat pipe. A valving arrangement was designed at the end of the condenser section to allow for fluid addition and vacuum connection. A schematic of the apparatus is shown in Fig. 8.

Prior to each experimental run, the heat pipe was evacuated to approximately 5 microns for a period of 24 hours to remove most of the foreign gases. The heat pipe was then charged with a 30 cubic centimeters of the appropriate water-ethanol solution. This amount was sufficient to saturate the entire wick structure. Electrical energy was added to the evaporator section to initiate the heat pipe cycle. For a given angle of inclination, the steady state temperature distribution in the vapor and at the wall of the heat pipe was measured. The above was

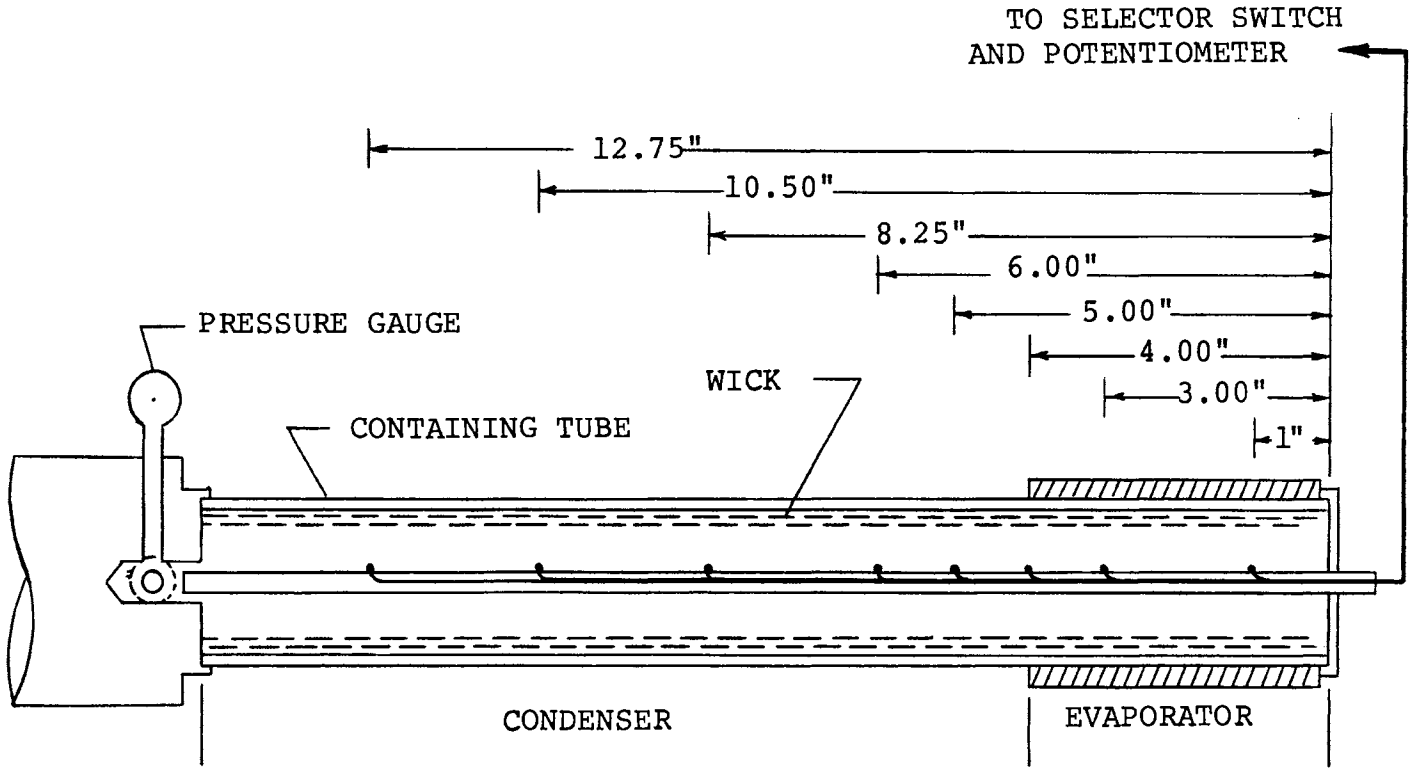

Fig. 7 Internal Thermocouple Location

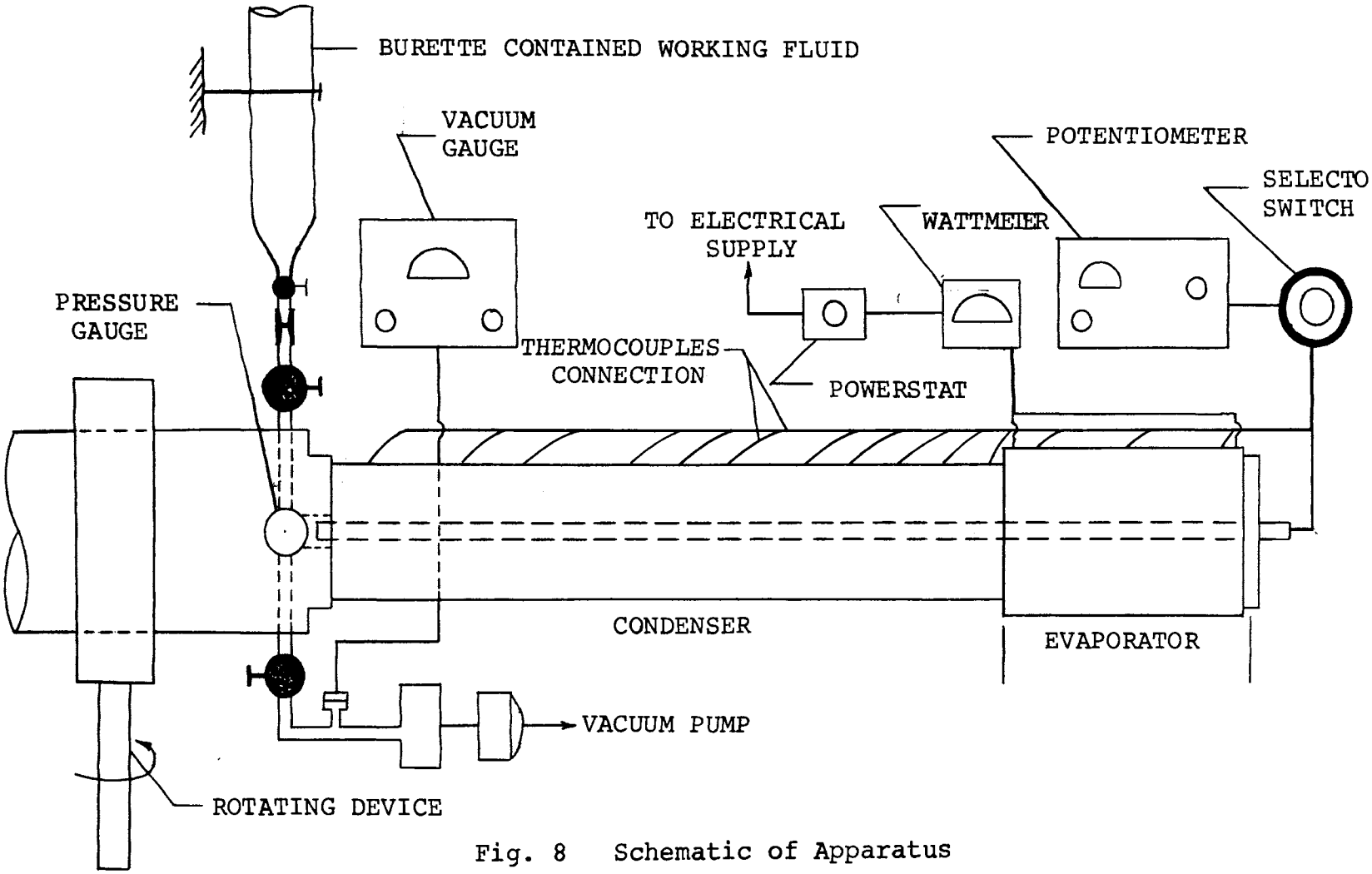

Fig. 8 Schematic of Apparatus

performed for two evaporator power loads 50 and 75 watts. The entire procedure was repeated for several water ethanol concentrations.

V. RESULTS AND DISCUSSION

Experimental evaluation of the two component heat pipe was performed using two power loads, 50 and 75 watts, and several concentration levels of water-ethanol mixture as the working fluid. The effect of inclination angle on its operating characteristic was determined. As overall experimental performance data consisting of the operating pressure, surface and vapor temperature distribution are summarized in Appendix B and C.

The performance of the heat pipe with a single component, water and ethanol, was performed first as a check and as a reference to the performance of the two component heat pipe. The surface temperature distribution for the 100% water and 100% ethanol heat pipe at a power load of 50 watts is shown in Fig. 9 and Fig. 10 respectively for various angles of inclination. Their performance exhibit the characteristics of a wick limited heat pipe. The performance of such a heat pipe is limited primarily by the ability of the wick to deliver liquid continuously and at a sufficient rate to the evaporator section. As can be seen for an angle of inclination beyond 3 degrees, when evaporator is above condenser, the temperature drop along the heat pipe length increases rapidly as an indication of partial drying out of the wick.

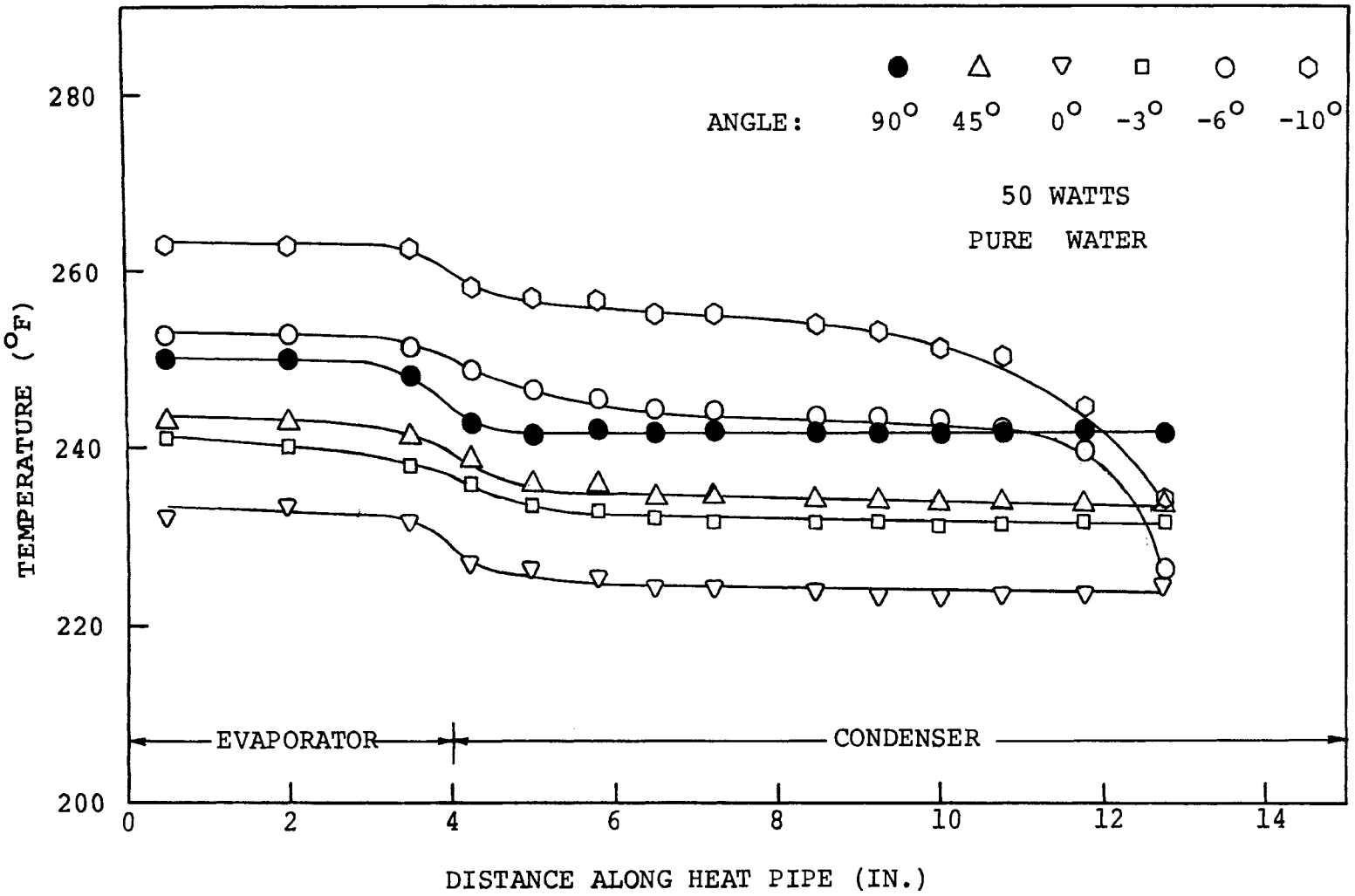

Fig. 9 Surface Temperature for Different Angles of Inclination 50 Watts, Pure Water

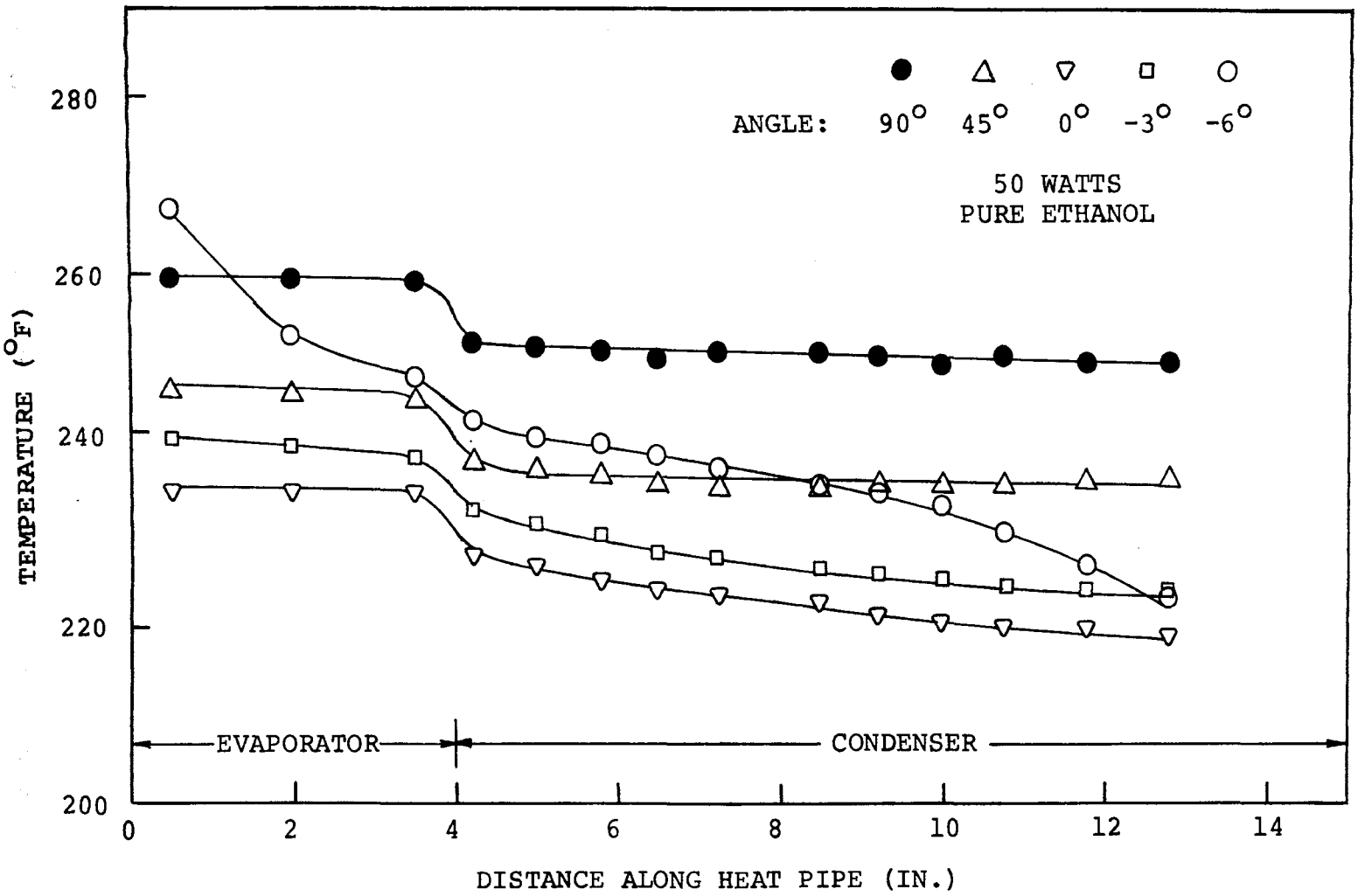

Fig. 10 Surface Temperature for Different Angles of Inclination, 50 Watts, Pure Ethanol

For the horizontal case, the water heat pipe experienced a $10^{o}F$ temperature drop along its length while operating at a pressure level of 1.54 atm. and the ethanol heat pipe experienced a drop of $16^{o}F$ at an operating pressure of 3.45 atm. for the same power load of 50 watts. The condenser temperature was $225^{o}F$ for the water heat pipe and $219^{o}F$ for the ethanol heat pipe. The higher temperature drop for the ethanol case is in agreement with what could be predicted and is due to the lower ethanol molar latent heat and its lower surface tension. This behavior is evident in all the experimental results. The latent heat, governs the amount of mass flow required through the wick for a given power load as indicated in equation (8). An increase in the capillary pumping head is required to increase the mass flow otherwise a larger drop in temperature will take place. It is interesting to note that the vapor temperature as shown in Appendix B is approximately constant for the horizontal operating case and corresponds to the saturated temperatures of the working fluid at the operating pressure. This agreement indicates that the heat pipe and the measuring instrument were operating satisfactorily.

At the higher power load (75 watts) the evaporator temperature increased substantially causing a large temperature drop along the heat pipe length for cases when evaporator is inclined beyond 3 degrees below the

condenser. This behavior is an indication of drying out of the wick and is due to the inability of the wick to deliver a sufficient rate of working liquid to the evaporator section. A plot of the temperature distribution versus angle of inclination at this power load for water and ethanol heat pipe is shown in Fig. 11 and Fig. 12. The results for the single component heat pipe indicates that for a given power load the operating pressure and the evaporator temperature are minimum when the heat pipe is in a horizontal position. Any angular displacement in either direction will cause these parameters to increase. This is due to the fact that when the heat pipe is inclined away from the horizontal the gravity force aids or obstructs the flow of the liquid. When it is aiding, the evaporator is below the condenser, the liquid moves rapidly through the wick to a puddle formed at the end of the evaporator section, leaving the remaining part of the wick less than completely saturated. Similarly when it is obstructing the flow, the evaporator is above the condenser, the wick is drying due to the limiting pumping capability of the wick. In the two cases wick dryness causes higher operating temperatures which require higher operating pressures.

The experimental results for the two component heat pipe exhibit the same behavior as the ones of Tien [20] for water-ethanol solution and those of Feldman [22] for water-

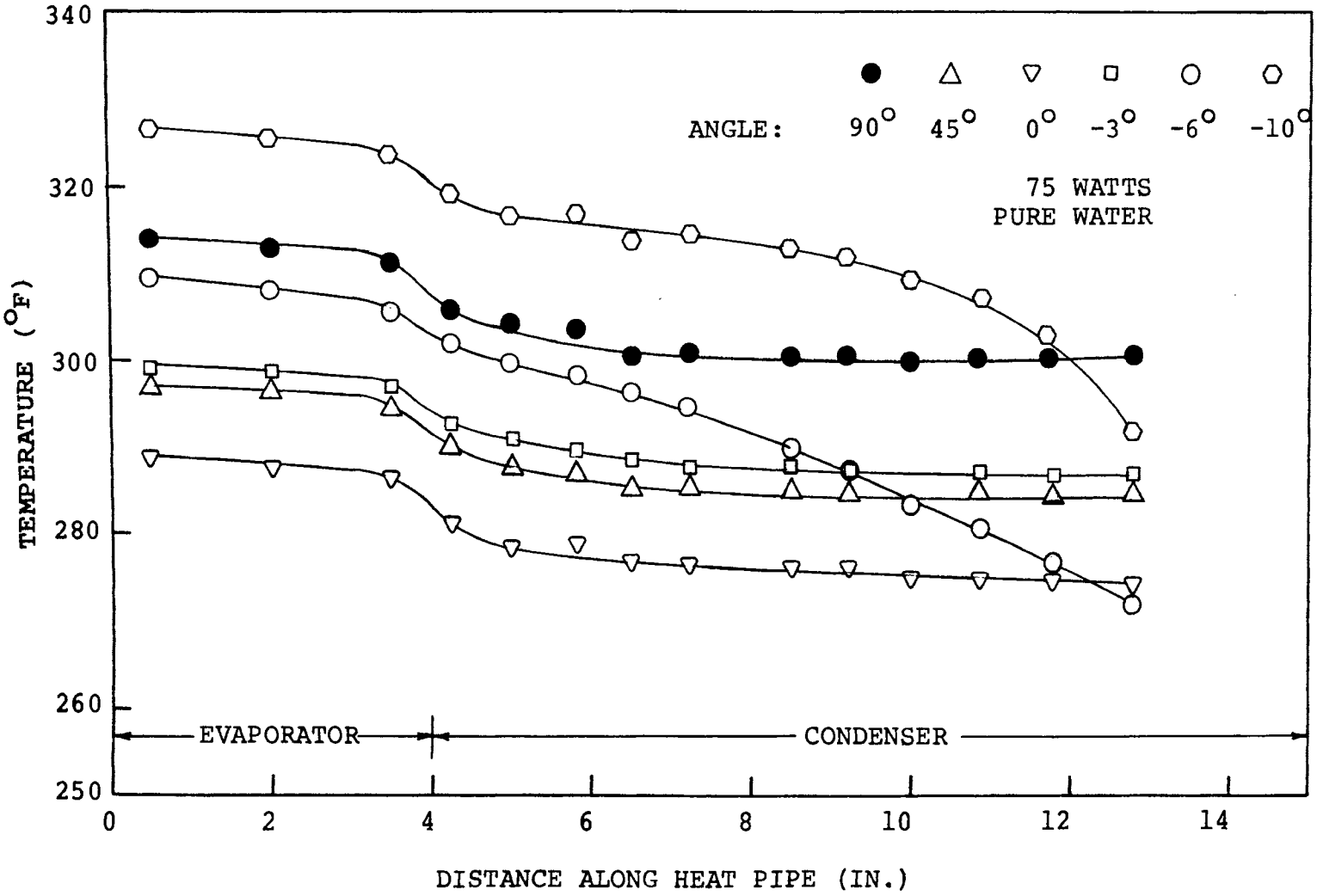

Fig. 11 Surface Temperature for Different Angles of Inclination, 75 Watts, Pure Water

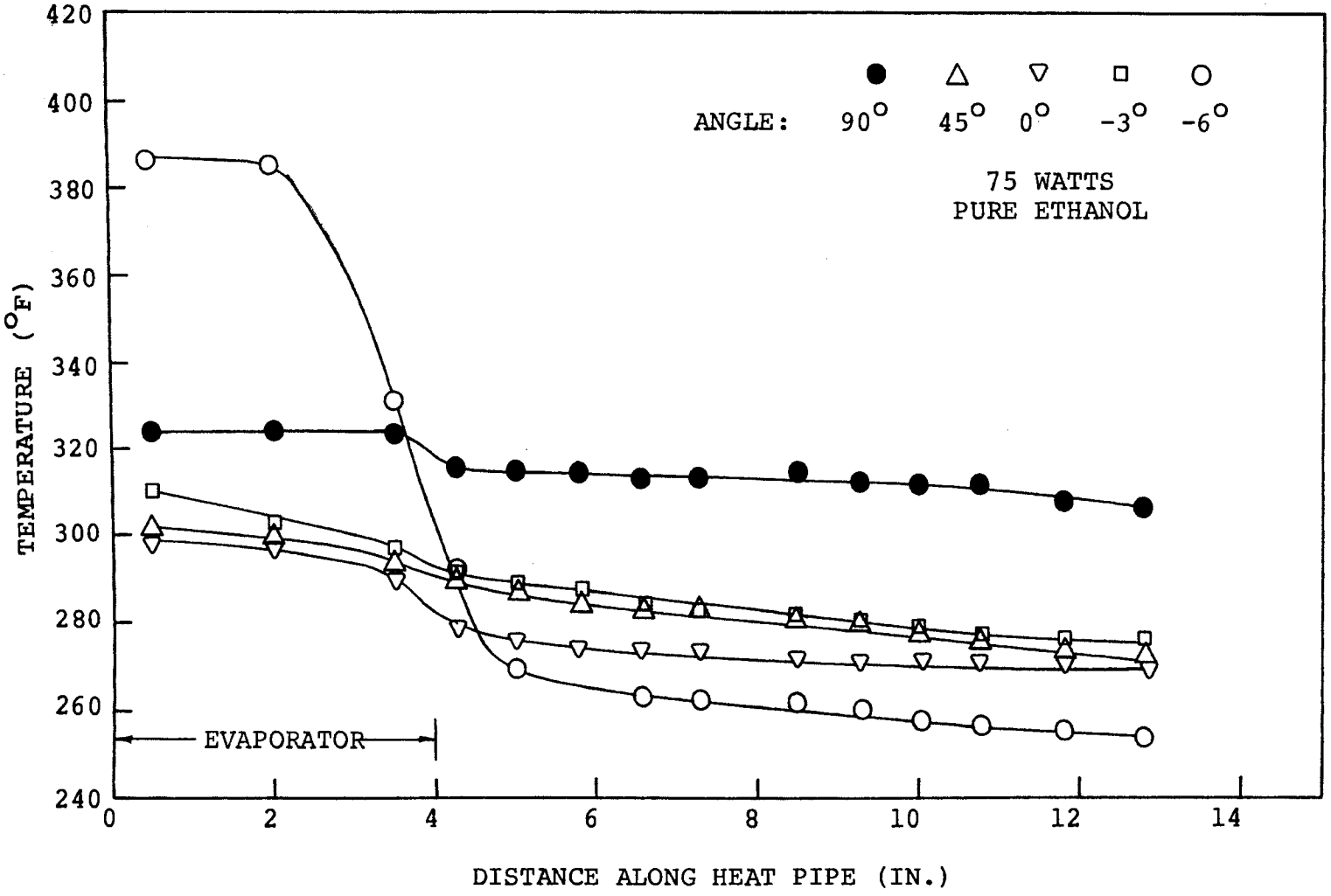

Fig. 12 Surface Temperature for Different Angles of Inclination, 75 Watts, Pure Ethanol

methanol solution. A comparison with their results could not be made since they operated at different power loads. The isothermal characteristic of the one component heat pipe do not exist in two component heat pipe. Instead, in most cases, two different isothermal regions will exist along the length of the heat pipe. The results for the horizontal operating condition are treated in detail in the remainder of this discussion. Data on its performance for different angles of inclination are tabulated in Appendix B and C.

The performance of the two component heat pipe with various levels of water-ethanol concentration is shown in Fig. 13 through 16. The surface and the vapor temperature are presented for the 50 watts and 75 watts power load. It is apparent that the temperature drop across the length of the heat pipe is greater than the drop for either of the single fluid heat pipe. The result can be characterized in general by a relatively constant high temperature region in the evaporator and similar low temperature region in the condenser. The differences in results between the one and the two component heat pipe indicate a possible difference in the mode of their operation.

The experimental results will be analyzed based on the two existing hypotheses. The first, formulated by Cotter [11], indicating that the fluids separate into relatively distinct regions, each operating at its own

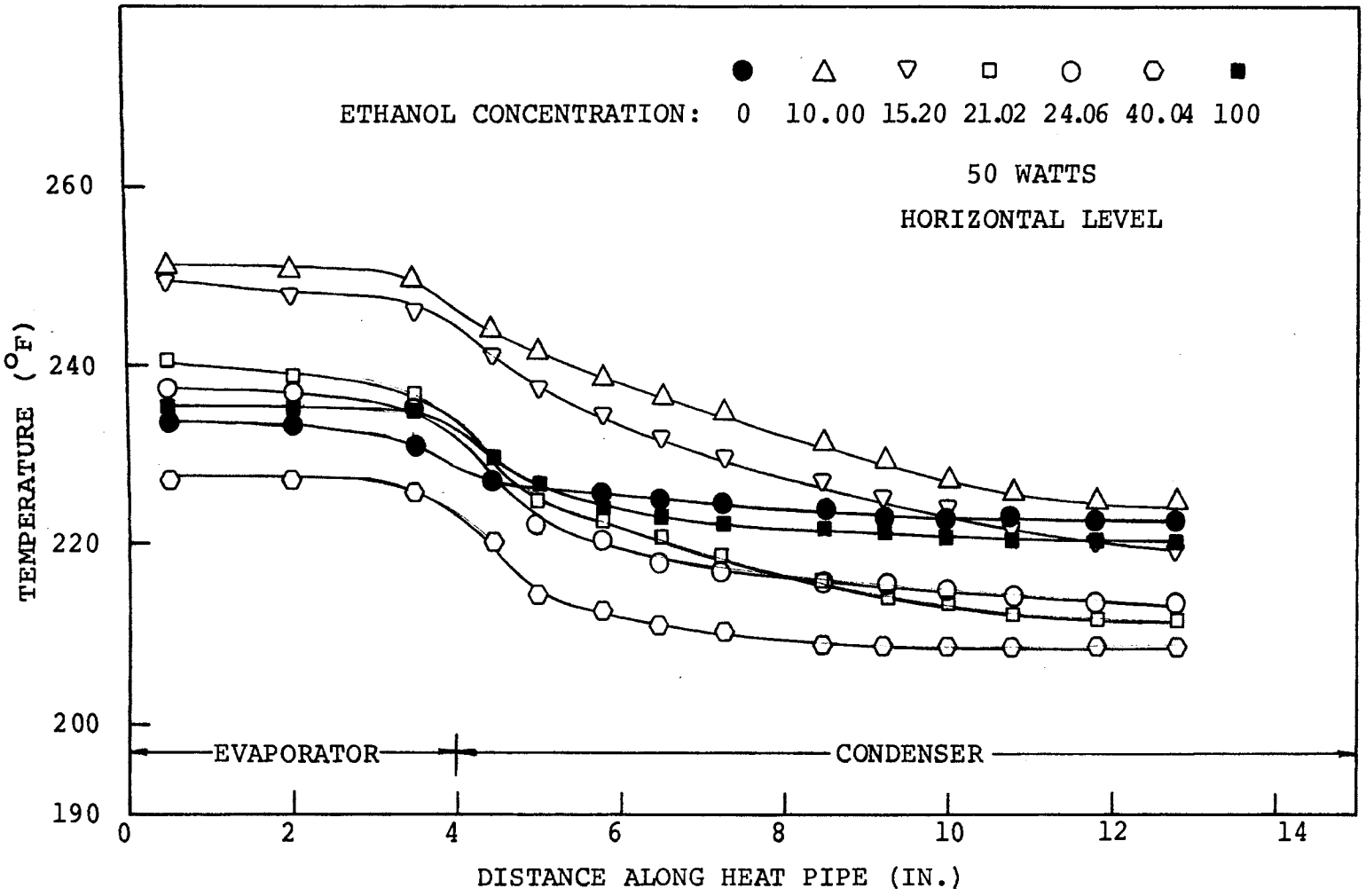

Fig. 13 Surface Temperature for Different Concentration, 50 Watts, Horizontal Level

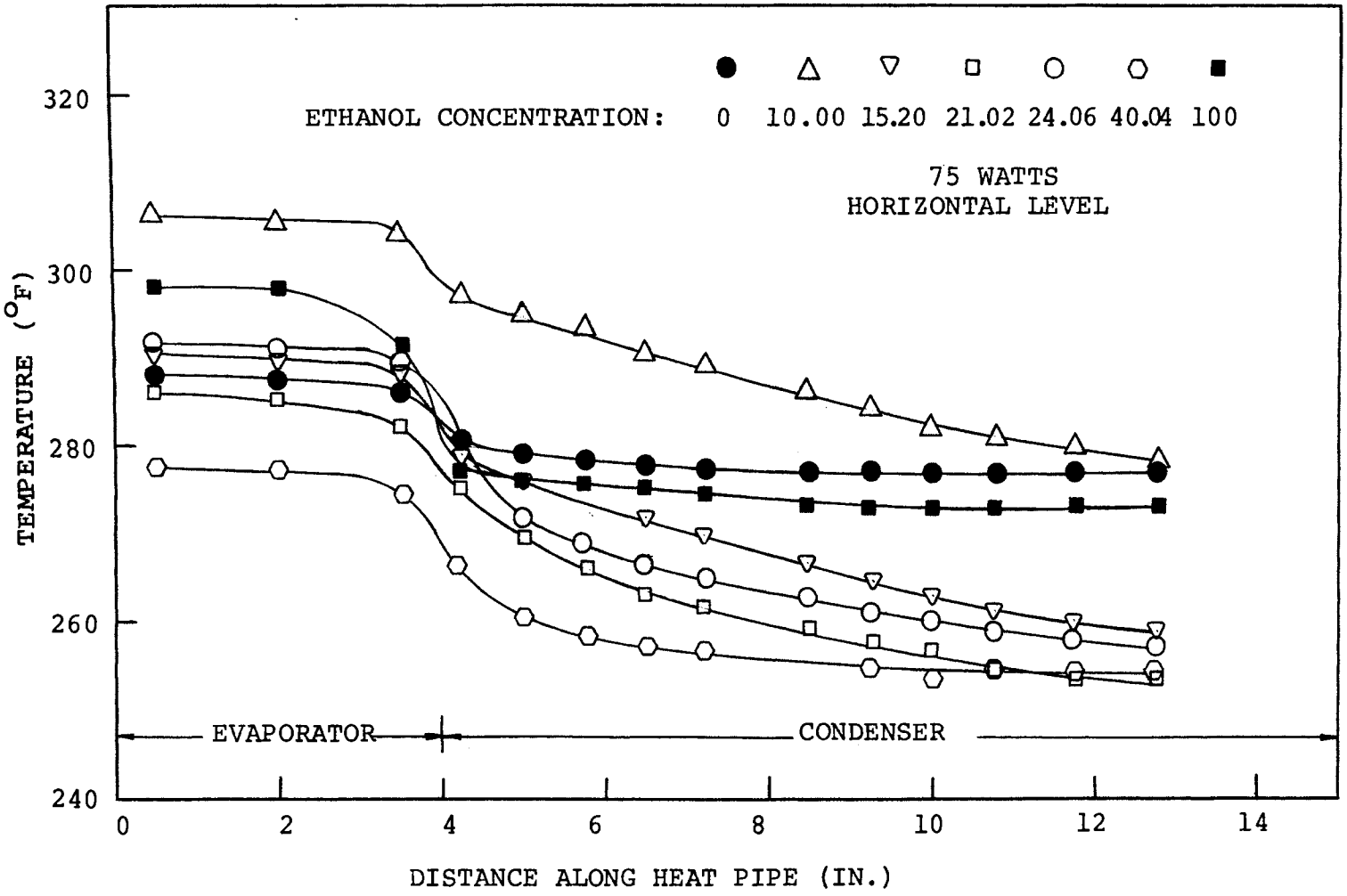

Fig. 14 Surface Temperature for Different Concentrations, 75 Watts, Horizontal Level

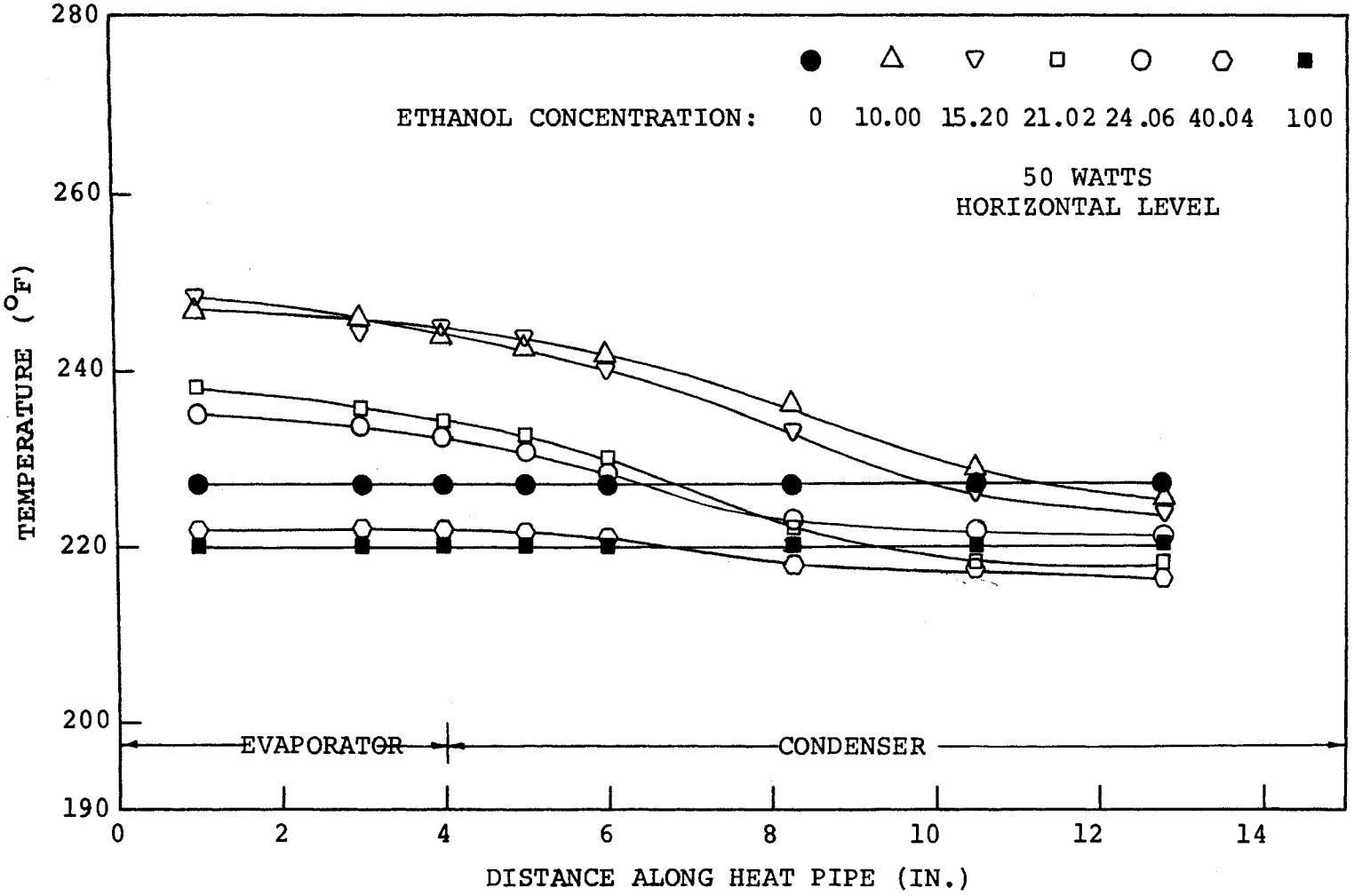

Fig. 15 Inside Temperature for Different Concentration, 50 Watts, Horizontal Level

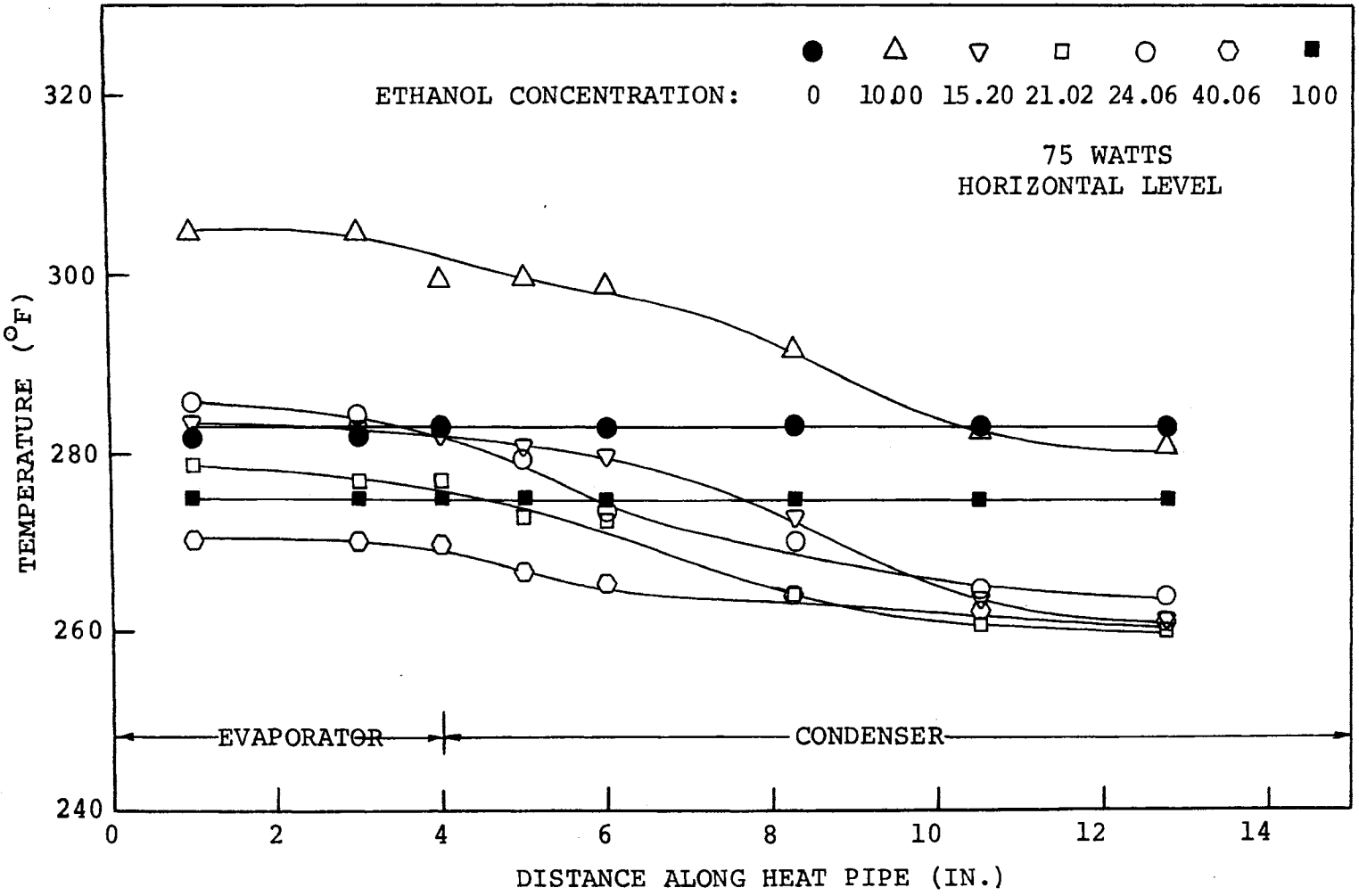

Fig. 16 Inside Temperature for Different Concentration, 75 Watts, Horizontal Level

saturated temperature. The second hypothesis, formulated by Tien [20], claims that separation of the two fluids will not take place and the phase equilibrium of the solution will govern the concentration of each component at any section of the heat pipe. Neither one of these have been proved or critically verified by an experiment.

The first hypothesis indicates that the fluid having the higher saturation temperature, in this case water, will occupy the higher temperature evaporator region. The other fluid having a lower saturation temperature will occupy the lower temperature condenser region. Heat then would transfer from higher to lower temperature region by conduction and mixing at the interface between the two regions. This hypothesis is evaluated by comparing the temperatures measured with the theoretical saturation vapor temperatures of the two fluids at the measured operating pressure. As shown in Table 1 and 2, large discrepancies exist between the measured and the saturated temperatures specially for the medium range of initial concentration. This fact indicates that the hypothesis of complete separation is not valid for the present experimental two component heat pipe.

The second hypothesis is evaluated by examining Fig. 17 and Fig. 18 where the experimental ethanol partial pressure in the evaporator section and the experimental water partial pressure in the condenser section are presented for different initial concentration. These figures clearly

Table 1 Experimental and Theoretical Results for Horizontal Level at 50 Watts

Run Number	Initial Conc.	Pressure (Atm.)	T_{ve} °F	T_{ws} °F	T_{vc} °F	T_{es} °F
1	0	1.54	227	227	227	194
2	5.05	1.92	237	242	228	200
3	10.00	1.82	246	244	226	202
4	15.20	1.82	245	250	224	202
5	21.02	1.95	234	250	218	205
6	24.06	2.09	234	252	201	209
7	30.00	2.97	244	273	234	228
8	35.04	3.04	243	274	235	229
9	40.04	2.29	223	257	217	214
10	45.31	2.10	217	252	213	210
11	50.16	2.56	225	265	223	220
12	60.30	2.50	219	262	220	220
13	70.10	3.72	240	287	224	240
14	80.62	3.86	238	289	236	242
15	91.00	3.38	226	281	228	234
16	100.00	3.45	220	282	220	235

Table 2 Experimental and Theroetical Results for Horizontal Level at 75 Watts

Run Number	Initial Conc.	Pressure (Atm.)	T_{ve} °F	T_{ws} °F	T_{vc} °F	T_{es} °F
1	0	3.86	282	289	284	242
2	5.05	4.06	288	292	275	246
3	10.00	5.08	304	307	281	258
4	15.20	3.73	282	287	262	240
5	21.02	4.20	277	294	261	248
6	24.06	4.74	284	302	265	254
7	30.00	7.25	298	332	287	281
8	35.04	6.58	293	325	280	275
9	40.04	5.08	270	307	262	259
10	45.31	5.15	270	308	265	260
11	50.16	5.56	273	313	269	264
12	60.30	9.10	296	350	293	296
13	70.10	8.25	288	342	283	290
14	80.62	8.00	282	340	283	287
15	91.00	8.00	280	340	283	287
16	100.00	9.06	280	341	278	297

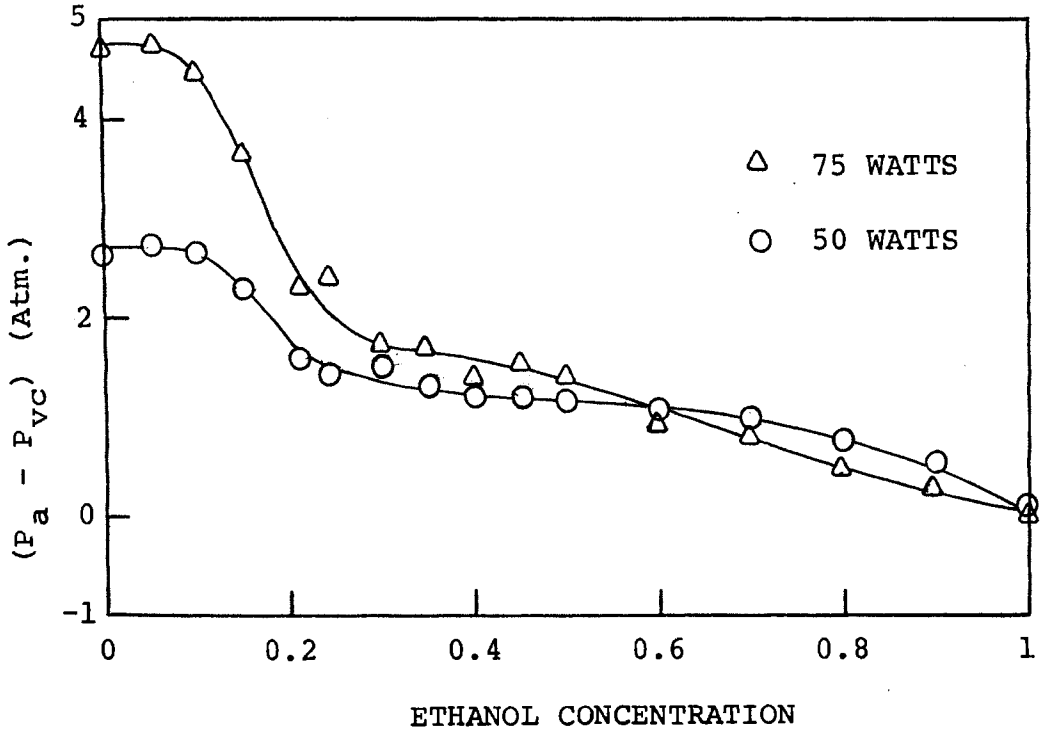

Fig. 17 Variation of Partial Pressure of Ethanol With Initial Concentration

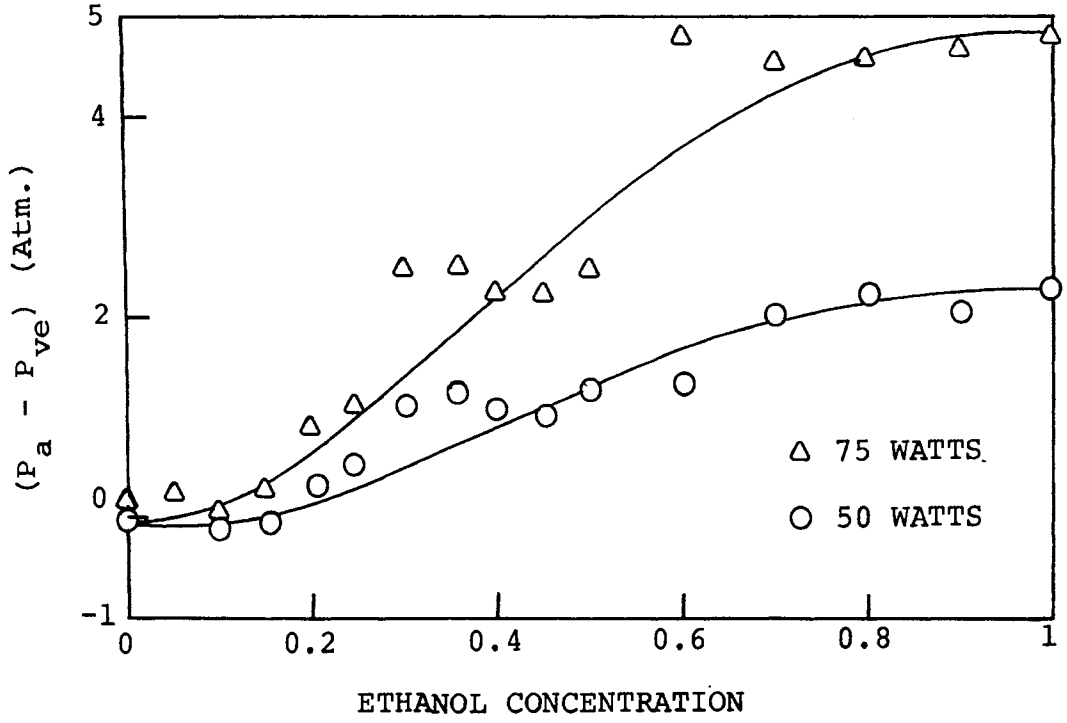

Fig. 18 Variation of Partial Pressure of Water With Initial Concentration

demonstrate that separation of the two fluids does not take place. For the case when the initial concentration is rich in water, almost pure water occupies the evaporator (partial pressure of ethanol is almost zero) but the condenser is filled with water ethanol mixture. When the initial concentration is rich in ethanol, almost pure ethanol occupies the condenser (partial pressure of water is almost zero) but the evaporator is filled with water ethanol mixture. The figures also indicate that for any other initial concentration complete separation will not take place. This, however, does not mean that the local concentration is governed by the thermodynamic phase equilibrium. Some more results are needed to validate this fact.

The axial temperature distribution appear to depend strongly on the initial concentration. The functional dependence is quite complicated and may depend on the properties of the solution and the wick. The evaporator temperature fluctuate as the ethanol concentration increases. The most sensitive region appears to exist within the range of initial concentrations 0-20 percent ethanol. It is interesting to note that the properties of water-ethanol solution behave strangely and in a unique way in this range of ethanol concentration. As can be seen from section III the latent heat reaches a maximum and the mass diffusion coefficient reaches a minimum [27]

in this range. Other properties are not available for this solution, however it is believed that the strange behavior will exist in most of the properties in this range of concentration. Further theoretical investigations regarding the properties of the solution and the behavior of the two component heat pipe are needed in order to predict the functional relation between the initial concentration and the temperature distribution.

The effect of the angle of inclination on the temperature distribution for a two component heat pipe is presented in Fig. 19 and Fig. 20 for the case of 10 percent ethanol. Appendix B and C contain data for other concentrations. The effect is similar to the case of a one component heat pipe which have been discussed earlier.

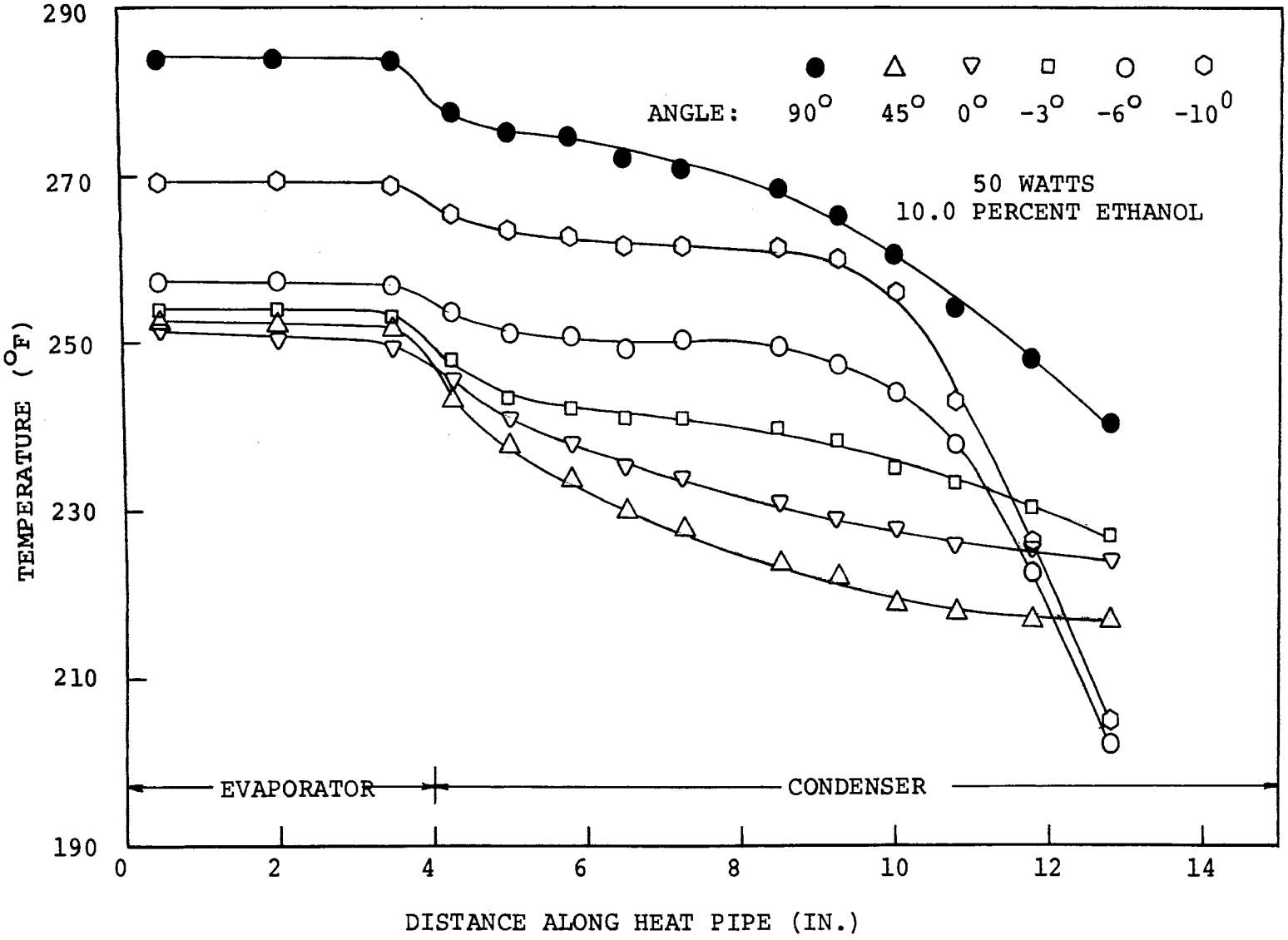

Fig. 19 Surface Temperature for Different Angles of Inclination, 50 Watts, 10.00 Percent Ethanol

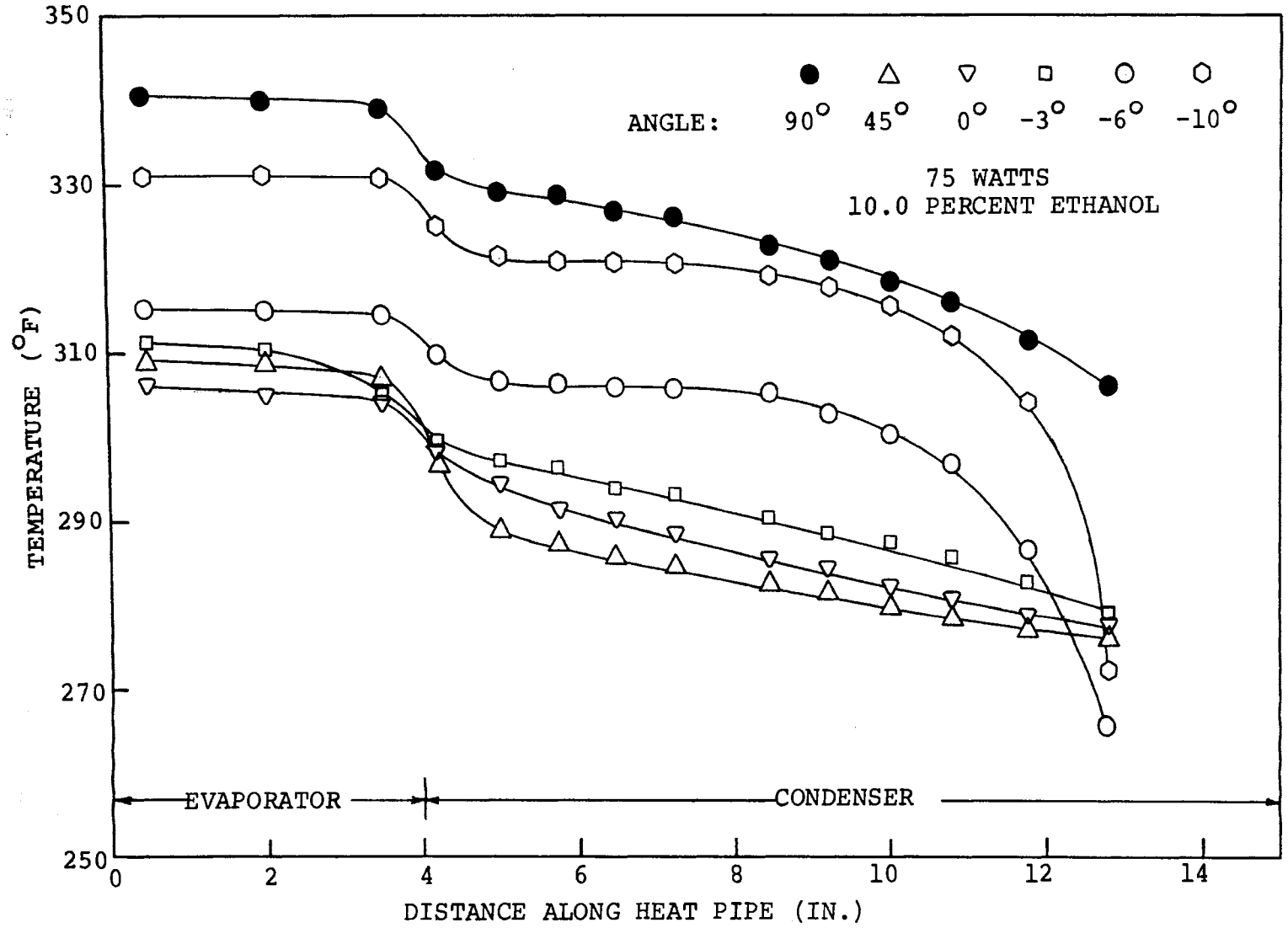

Fig. 20 Surface Temperature for Different Angles of Inclination, 75 Watts, 10.00 Percent Ethanol

VI. CONCLUSIONS AND RECOMMENDATIONS

The operating characteristics of a two-component heat pipe have been demonstrated. The results indicate that in most cases complete separation of the two components does not take place in the heat pipe. At low ethanol concentration the evaporator contains almost pure water while the condenser contains a mixture of water and ethanol. At high ethanol concentration the condenser contains pure ethanol while the evaporator contains a mixture of water and ethanol. The effect of inclination angle on the temperature distribution of a two component heat pipe is similar to the case of the one component heat pipe. Wick dryness starts to take place at angles beyond 3 degrees with the condenser below evaporator.

It is recommended that further experimentation be made with a Pyrex glass heat pipe, to permit some visual aid, for the purpose of examining the flow patterns within the heat pipe. Sampling stations, where fluid can be extracted and chemically analyzed, should be installed in order to examine if the phase equilibrium actually governs the concentration of the solution at different sections of the heat pipe. It is essential to perform experiments with other types of fluids since the properties of water-ethanol solutions behave in some what strange fashion at low

concentration. The additional results would help to formulate the functional relationship between the fluid properties and the axial temperature distribution of the heat pipe.

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VITA

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He entered Chulalongkorn University, Bangkok, in 1962 and received the Diploma of Science in Bio-Chemistry, in 1965. In 1968 he received the Bachelor of Science degree in Mechanical Engineering.

He came to the United States to the University of Kansas in September 2, 1969 and transferred to the University of Missouri-Rolla in the Fall of 1970.

APPENDIX A
PROPERTIES OF ETHANOL

Temp. Cent. t	Temp. Fahr. T	Press. Atm. P	Enthalpy (Btu/lb) Sat. Liquid h_f	Latent Heat h_{fg}	Sat. Vapor h_g
26	78.80	0.08	25.39	401.55	426.94
27	80.60	0.09	26.44	400.94	427.38
28	82.40	0.09	27.49	400.33	427.82
29	84.20	0.10	28.55	399.73	428.27
30	86.00	0.10	29.61	399.12	428.73
31	87.80	0.11	30.67	398.52	429.20
32	89.60	0.11	31.75	397.92	429.67
33	91.40	0.12	32.82	397.33	430.15
34	93.20	0.13	33.91	396.73	430.64
35	95.00	0.14	35.00	396.14	431.13
36	96.80	0.14	36.09	395.54	431.63
37	98.60	0.15	37.19	394.95	432.14
38	100.40	0.16	38.29	394.36	432.65
39	102.20	0.17	39.40	393.77	433.17
40	104.00	0.18	40.52	393.18	433.70
41	105.80	0.19	41.63	392.59	434.23
42	107.60	0.20	42.76	392.00	434.76
43	109.40	0.21	43.89	391.42	435.31
44	111.20	0.22	45.02	390.83	435.85
45	113.00	0.23	46.17	390.24	436.41
46	114.80	0.24	47.31	389.66	436.97
47	116.60	0.25	48.46	389.07	437.53
48	118.40	0.26	49.62	388.48	438.10
49	120.20	0.28	50.78	387.89	438.67
50	122.00	0.29	51.95	387.30	439.25

APPENDIX A (continued)

Temp. Cent. t	Temp. Fahr. T	Press. Atm. P	Enthalpy (Btu/lb) Sat. Liquid h_f	Latent Heat h_{fg}	Sat. Vapor h_g
51	123.80	0.30	53.12	386.71	439.83
52	125.60	0.32	54.30	386.13	440.42
53	127.40	0.34	55.48	385.53	441.01
54	129.20	0.35	56.67	384.94	441.61
55	131.00	0.37	57.86	384.35	442.21
56	132.80	0.39	59.06	383.76	442.81
57	134.60	0.40	60.26	383.16	443.42
58	136.40	0.42	61.47	382.56	444.03
59	138.20	0.44	62.68	381.97	444.65
60	140.00	0.46	63.90	381.37	445.27
61	141.80	0.48	65.13	380.76	445.89
62	143.60	0.51	66.36	380.16	446.51
63	145.40	0.53	67.59	379.55	447.14
64	147.20	0.55	68.83	378.94	447.78
65	149.00	0.58	70.08	378.33	448.41
66	150.80	0.60	71.33	377.72	449.05
67	152.60	0.63	72.58	377.10	449.69
68	154.40	0.66	73.84	376.49	450.33
69	156.20	0.68	75.11	375.86	450.98
70	158.00	0.71	76.38	375.24	451.62
71	159.80	0.74	77.66	374.61	452.27
72	161.60	0.77	78.94	373.98	452.92
73	163.40	0.81	80.23	373.35	453.58
74	165.20	0.84	81.52	372.71	454.23
75	167.00	0.88	82.82	372.07	454.89
76	168.80	0.91	84.13	371.42	455.55
77	170.60	0.95	85.44	370.77	456.21
78	172.40	0.99	86.75	370.12	456.87
79	174.20	1.03	88.07	369.46	457.53

APPENDIX A (continued)

Temp. Cent. t	Temp. Fahr. T	Press. Atm. P	Enthalpy (Btu/lb) Sat. Liquid h_f	Latent Heat h_{fg}	Sat. Vapor h_g
80	176.00	1.07	89.39	368.80	458.19
81	177.80	1.11	90.72	368.13	458.86
82	179.60	1.15	92.06	367.46	459.52
83	181.40	1.20	93.40	366.79	460.19
84	183.20	1.25	94.75	366.11	460.85
85	185.00	1.29	96.10	365.42	461.52
86	186.80	1.34	97.46	364.73	462.19
87	188.60	1.40	98.82	364.04	462.85
88	190.40	1.45	100.18	363.34	463.52
89	192.20	1.50	101.56	362.63	464.19
90	194.00	1.56	102.94	361.92	464.85
91	195.80	1.62	104.32	361.20	465.52
92	197.60	1.68	105.71	360.48	466.18
93	199.40	1.74	107.10	359.75	466.85
94	201.20	1.80	108.50	359.01	467.51
95	203.00	1.87	109.91	358.27	468.18
96	204.80	1.93	111.32	357.52	468.24
97	206.60	2.00	112.73	356.77	469.50
98	208.40	2.07	114.15	356.01	470.16
99	210.20	2.15	115.58	355.24	470.82
100	212.00	2.22	117.01	354.46	471.47
101	213.80	2.30	118.45	353.68	472.13
102	215.60	2.38	119.89	352.89	472.78
103	217.40	2.46	121.34	352.10	473.43
104	219.20	2.55	122.79	351.29	474.08
105	221.00	2.63	124.25	350.48	474.73
106	222.80	2.72	125.71	349.66	475.37
107	224.60	2.81	127.18	348.83	476.01
108	226.40	2.91	128.65	348.00	476.65

APPENDIX A (continued)

Temp. Cent. t	Temp. Fahr. T	Press. Atm. P	Enthalpy (Btu/lb) Sat. Liquid h_f	Latent Heat h_{fg}	Sat. Vapor h_g
109	228.20	3.00	130.13	347.16	477.29
110	230.00	3.10	131.62	346.30	477.92
111	231.80	3.20	133.11	345.44	478.55
112	233.60	3.31	134.60	344.58	479.18
113	235.40	3.42	136.10	343.70	479.80
114	237.20	3.53	137.61	342.81	480.42
115	239.00	3.64	139.12	341.92	481.04
116	240.80	3.75	140.64	341.01	481.65
117	242.60	3.87	142.16	340.10	482.26
118	244.40	3.99	143.69	339.18	482.87
119	246.20	4.12	145.22	338.25	483.47
120	248.00	4.25	146.76	337.30	484.07
121	249.80	4.38	148.31	336.35	484.66
122	251.60	4.51	149.85	335.39	485.25
123	253.40	4.65	151.41	334.40	485.83
124	255.20	4.79	152.97	333.44	486.41
125	257.00	4.93	154.54	332.44	486.98
126	258.80	5.08	156.11	331.44	487.55
127	260.60	5.23	157.68	330.43	488.11
128	262.40	5.39	159.26	329.40	488.67
129	264.20	5.55	160.85	328.37	489.22
130	266.00	5.71	162.44	327.32	489.76
131	267.80	5.88	164.04	326.26	490.30
132	269.60	6.05	165.65	325.19	490.83
133	271.40	6.22	167.25	324.11	491.36
134	273.20	6.40	168.87	323.01	491.88
135	275.00	6.58	170.49	321.91	492.39
136	276.80	6.77	172.11	320.79	492.90
137	278.60	6.96	173.74	319.66	493.40

APPENDIX A (continued)

Temp. Cent. t	Temp. Fahr. T	Press. Atm. P	Enthalpy (Btu/lb) Sat. Liquid h_f	Latent Heat h_{fg}	Sat. Vapor h_g
138	280.40	7.15	175.38	318.51	493.89
139	282.20	7.35	177.02	317.36	494.38
140	284.00	7.55	178.67	316.19	494.85
141	285.80	7.76	180.32	315.01	495.32
142	287.60	7.97	181.98	313.81	495.79
143	289.40	8.19	183.64	312.60	496.24
144	291.20	8.41	185.31	311.38	496.68
145	293.00	8.64	186.98	310.14	497.12
146	294.20	8.87	188.66	308.89	497.55
147	296.60	9.11	190.35	307.62	497.97
148	298.40	9.35	192.04	306.34	498.38
149	300.20	9.59	193.73	305.05	498.78
150	302.00	9.84	195.43	303.74	499.17

APPENDIX B

EXPERIMENTAL RESULTS FOR THE HORIZONTAL AND VERTICAL MODE OF OPERATION AT VARIOUS CONCENTRATION

Table B-1 Deduced Experimental Results, Horizontal Level, 50 Watts

Run Number	Initial Conc.	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	0	1.54	234	234	232	227	227	227	223	225	225	223
2	5.05	1.92	240	240	240	236	234	233	231	231	231	229
3	10.00	1.82	251	250	249	244	241	238	235	235	234	230
4	15.20	1.82	250	248	246	242	237	235	231	231	229	225
5	21.02	1.95	241	238	236	230	226	223	223	223	218	214
6	24.06	2.09	238	238	235	230	222	221	218	218	218	216
7	30.00	2.97	249	250	246	233	232	228	228	228	227	225
8	35.04	3.04	248	248	245	249	232	230	228	228	228	226
9	40.04	2.29	227	228	226	221	215	214	213	211	212	209
10	45.31	2.10	221	220	219	208	200	205	203	204	203	200
11	50.16	2.56	232	231	230	211	209	212	211	212	209	211
12	60.30	2.50	228	227	226	208	208	208	208	212	206	206
13	70.10	3.72	246	248	244	231	226	228	228	228	204	227
14	80.62	3.86	245	248	245	232	226	228	227	229	229	226
15	91.00	3.38	234	235	234	221	223	227	228	228	228	216
16	100.00	3.45	235	235	235	228	228	224	224	224	224	221

Table B-1 (continued)

Run Number	Ext. Surface Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	223	224	224	225	227	227	227	227	227	227	227	227
2	226	226	226	224	237	237	237	237	237	236	230	228
3	229	224	224	224	247	246	244	243	242	236	229	226
4	221	221	220	219	248	245	245	243	240	233	236	224
5	212	212	212	238	238	234	234	233	230	222	218	218
6	214	214	214	214	235	234	233	231	228	203	201	201
7	225	225	225	224	244	244	243	240	237	235	234	234
8	225	225	225	224	243	243	242	241	234	236	235	235
9	207	210	210	210	222	223	222	223	221	218	218	217
10	201	201	200	201	217	217	217	217	216	214	213	213
11	209	205	197	200	225	225	225	225	224	224	223	223
12	207	204	205	205	219	219	222	222	219	220	220	220
13	227	226	209	200	240	240	241	240	240	240	239	224
14	226	222	216	206	237	238	238	238	238	238	237	236
15	220	215	218	217	226	226	227	227	228	228	228	228
16	220	220	221	219	220	220	220	220	220	220	220	220

Table B-2 Deduced Experimental Results Horizontal Level, 75 Watts

Run Number	Initial Conc.	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	0	3.86	288	287	286	281	279	280	277	277	277	277
2	5.05	4.06	295	293	294	288	284	284	280	283	282	279
3	10.00	5.08	306	305	304	297	295	294	290	290	290	285
4	15.20	3.73	290	289	288	279	276	276	273	269	267	263
5	21.02	4.20	286	285	282	275	269	266	262	262	262	287
6	24.06	4.74	292	291	289	281	272	270	266	266	265	262
7	30.00	7.25	307	305	298	294	286	286	281	281	281	278
8	35.04	6.58	298	297	291	287	280	279	275	275	276	272
9	40.04	5.08	276	278	274	267	261	259	258	259	258	255
10	45.31	5.15	277	275	273	253	253	253	250	252	219	249
11	50.16	5.56	281	278	276	257	254	254	252	254	224	253
12	60.30	9.10	307	310	302	282	280	283	279	282	280	280
13	70.10	8.25	298	302	294	281	275	277	277	278	280	274
14	80.62	8.00	289	290	286	275	269	269	270	272	272	268
15	91.00	8.00	285	283	283	283	268	268	269	269	267	265
16	100.00	9.06	298	298	291	277	276	276	277	275	273	273

Table B-2 (continued)

Run Number	Ext. Surface Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	274	274	274	274	282	282	283	283	283	284	284	284
2	274	274	274	274	288	288	288	288	288	287	280	275
3	280	279	278	278	304	304	300	300	295	292	283	281
4	260	261	261	259	283	282	282	281	280	273	263	262
5	253	253	254	254	279	277	277	275	272	264	261	261
6	258	258	258	258	286	284	283	280	273	271	265	265
7	277	277	278	277	300	298	298	294	291	289	287	287
8	269	270	273	270	294	293	290	289	285	282	280	280
9	253	254	254	254	270	271	270	267	266	264	263	262
10	248	243	256	256	270	270	270	269	268	266	265	265
11	253	248	260	259	273	273	273	272	270	269	269	269
12	279	278	281	270	296	296	296	296	296	297	296	293
13	272	272	272	251	288	288	290	289	289	289	288	283
14	268	263	263	256	282	282	284	284	284	285	285	283
15	265	260	260	260	280	280	282	283	283	283	283	283
16	263	273	273	273	280	280	278	278	278	278	280	278

Table B-3 Deduced Experimental Results Vertical Level, 50 Watts

Run Number	Initial Conc.	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	0	2.16	250	250	248	242	241	243	242	243	244	244
2	5.05	2.26	256	255	255	248	248	248	245	245	245	244
3	10.00	3.72	284	284	284	277	275	275	269	269	269	266
4	15.20	3.72	281	280	279	271	269	257	246	244	243	239
5	21.02	4.40	286	285	283	268	258	246	245	248	245	243
6	24.06	3.72	274	273	271	254	239	239	236	236	236	236
7	30.00	4.33	272	272	268	258	247	246	246	243	246	244
8	35.04	5.08	271	271	268	251	246	246	246	246	246	246
9	40.04	3.38	247	246	244	235	225	225	224	226	220	224
10	45.31	3.65	246	245	244	227	221	221	218	218	218	215
11	50.16	2.97	236	235	234	214	211	213	214	213	213	212
12	60.30	3.72	248	245	245	234	226	226	225	225	222	223
13	100.00	5.15	259	259	259	251	252	252	250	253	252	250

Table B-3 (continued)

Run Number	Ext. Surface Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	244	244	244	244	247	248	249	249	249	250	250	250
2	241	241	241	236	251	251	251	251	253	253	251	243
3	261	252	248	244	280	280	281	281	281	280	280	259
4	238	237	237	236	277	278	278	274	271	268	250	243
5	241	243	243	243	279	279	278	267	266	258	253	252
6	236	237	238	238	269	269	268	259	254	249	247	246
7	244	247	247	247	268	268	267	262	259	258	257	256
8	247	247	247	247	268	268	267	262	259	258	257	256
9	223	226	227	227	241	241	241	233	233	233	233	233
10	215	214	218	218	240	240	241	234	234	234	234	234
11	214	209	213	213	233	233	234	229	229	229	229	229
12	223	220	225	224	242	241	243	239	239	239	239	239
13	249	252	250	250	256	256	257	258	259	260	260	260

Table B-4 Deduced Experimental Results, Vertical Level, 75 Watts

Run Number	Initial Conc.	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	0	5.84	314	312	311	306	305	305	300	301	302	302
2	5.05	6.45	321	319	319	313	311	311	309	310	308	308
3	10.00	7.88	341	339	339	331	329	330	325	327	322	320
4	15.20	6.70	319	316	315	305	295	293	282	283	282	278
5	21.02	9.84	338	339	337	321	297	295	290	292	293	291
6	24.06	6.57	309	307	305	290	290	270	267	270	268	265
7	30.00	10.88	334	332	330	313	297	296	294	294	295	294
8	35.04	11.05	334	333	327	314	303	303	302	302	305	303
9	40.04	7.80	304	302	299	288	277	277	275	278	275	272
10	45.31	11.00	325	322	321	300	289	292	290	294	289	292
11	50.16	8.95	309	305	305	283	281	280	276	276	275	273
12	60.30		345	347	340	338	318	321	321	318	318	317
13	100.00	14.00	324	324	324	315	315	315	313	314	315	312

Table B-4 (continued)

Run Number	Ext. Surface Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	300	301	301	301	309	310	311	312	312	313	313	313
2	308	307	307	307	316	317	319	319	319	319	317	311
3	315	315	311	306	331	331	332	332	332	332	326	314
4	274	272	275	272	331	312	313	295	286	281	277	276
5	290	290	290	291	334	335	334	317	315	306	302	300
6	264	266	268	268	300	301	301	300	300	312	311	311
7	294	295	295	295	318	329	329	317	314	310	308	307
8	301	302	302	302	328	329	322	318	317	315	314	314
9	273	270	270	270	294	295	294	283	283	283	282	282
10	293	293	303	301	318	318	315	311	311	310	310	310
11	275	272	285	284	298	298	298	290	290	289	289	289
12	317	320	320	320	336	336	336	336	336	336	336	336
13	312	312	307	306	319	320	320	322	324	324	324	324

APPENDIX C

EXPERIMENTAL RESULTS FOR DIFFERENT CONCENTRATIONS AND ANGLES OF INCLINATION

Table C-1 Deduced Experimental Results, Pure Water, 50 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	2.16	250	250	248	242	241	243	242	243	244	244
2	45	1.75	243	243	241	239	235	236	234	235	235	234
3	0	1.34	234	234	232	227	227	227	223	225	225	223
4	-3	1.68	241	241	238	236	234	235	234	232	233	232
5	-6	2.19	253	253	251	249	247	247	244	244	245	244
6	-10	2.56	263	263	262	258	257	258	254	255	255	253

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	244	244	244	244	247	248	249	249	249	250	250	250
2	234	234	234	234	236	236	237	237	238	238	238	238
3	223	224	224	225	227	227	227	227	227	227	227	227
4	232	231	233	232	235	235	235	235	236	236	236	236
5	241	242	240	226	247	247	247	247	247	247	247	204
6	250	252	242	229	258	258	252	252	252	252	252	198

Table C-2 Deduced Experimental Results, Pure Water, 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	5.84	314	312	311	306	305	305	300	301	302	302
2	45	4.40	297	296	294	290	288	288	285	286	286	286
3	0	3.86	288	287	286	281	279	280	277	277	277	277
4	-3	4.47	299	298	295	291	289	291	286	287	288	286
5	-6	5.35	310	307	306	302	301	300	296	296	300	296
6	-10	6.90	326	325	323	319	316	319	312	315	313	313

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	300	301	301	301	309	310	311	312	312	313	313	313
2	282	285	285	285	290	292	292	293	293	293	293	293
3	274	274	274	274	282	282	283	283	283	284	284	284
4	282	285	286	286	292	292	293	293	293	293	293	290
5	293	293	287	282	303	304	305	305	305	306	306	263
6	309	308	304	290	322	322	322	322	322	322	317	269

Table C-3 Deduced Experimental Results, 5.05 Percent Ethanol Concentration, 50 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	2.26	256	255	255	248	248	248	245	245	245	244
2	45	1.92	248	246	245	241	239	239	237	238	238	236
3	0	1.92	240	240	240	236	234	233	231	231	231	229
4	-3	1.95	246	246	245	241	238	238	235	236	237	235
5	-6	2.19	253	253	252	248	246	246	243	245	245	243
6	-10	2.70	266	266	265	262	259	259	257	257	257	257

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	241	241	241	236	251	251	251	251	253	253	251	243
2	230	230	230	229	242	242	242	242	243	240	229	230
3	226	226	226	224	237	237	237	237	237	236	230	228
4	233	232	232	230	240	241	242	242	242	242	234	231
5	239	239	238	228	248	248	249	249	249	249	245	221
6	255	255	250	234	261	262	262	262	263	263	260	220

Table C-4 Deduced Experimental Results, 5.05 Percent Ethanal Concentration, 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	6.45	321	319	319	313	311	311	309	310	308	308
2	45	4.75	300	299	298	291	289	289	286	287	287	284
3	0	4.06	295	293	294	288	284	284	280	283	282	279
4	-3	4.20	297	294	294	288	285	286	283	284	284	280
5	-6	5.00	302	301	301	298	296	297	296	296	246	293
6	-10	6.45	322	322	321	317	313	313	312	313	313	312

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	308	307	307	307	316	317	319	319	319	319	317	311
2	278	278	278	272	295	296	294	292	291	287	281	283
3	274	274	274	274	288	288	288	288	288	287	280	275
4	278	278	278	276	289	290	291	291	291	291	284	277
5	292	292	289	278	301	302	302	303	303	303	299	268
6	307	308	306	289	323	323	322	318	313	314	308	312

Table C-5 Deduced Experimental Results, 10.00 Percent Ethanol Concentration, 50 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	3.72	284	284	284	277	275	275	269	269	269	266
2	45	2.26	253	252	251	243	238	234	230	230	227	223
3	0	1.82	251	250	249	244	241	238	235	235	234	230
4	-3	2.09	254	254	253	244	242	242	239	241	242	239
5	-6	2.36	257	256	256	254	251	251	248	251	250	247
6	-10	2.87	269	269	269	266	263	263	263	262	263	261

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	261	252	248	240	280	280	281	281	281	280	280	259
2	217	217	219	217	248	248	248	242	241	227	223	222
3	229	224	224	224	247	246	244	243	242	236	229	226
4	234	233	228	218	253	253	250	250	250	248	238	217
5	244	242	216	202	254	254	254	255	255	255	244	196
6	258	243	224	202	265	266	267	267	267	267	262	197

Table C-6 Deduced Experimental Results, 10.00 Percent Ethanol Concentration, 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	7.88	341	339	339	331	329	330	325	327	322	320
2	45	5.70	309	309	307	297	289	290	285	287	284	283
3	0	5.08	306	305	304	297	245	294	290	290	240	285
4	-3	5.22	310	310	304	299	297	297	296	295	294	291
5	-6	5.97	315	315	315	310	307	307	306	306	307	303
6	-10	7.47	331	331	331	325	321	321	320	321	320	318

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	315	315	311	306	331	331	332	332	332	332	326	214
2	277	277	277	278	303	303	299	297	293	286	286	284
3	280	279	278	278	304	304	300	300	295	292	283	281
4	285	283	280	267	300	300	301	301	301	299	287	268
5	299	297	287	266	311	311	312	312	313	313	301	262
6	315	313	304	272	326	326	326	326	320	320	312	255

Table C-7 Deduced Experimental Results, 15.20 Percent Ethanol Concentration, 50 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	3.72	281	280	279	271	262	257	246	244	243	239
2	45	2.50	257	257	255	243	229	225	222	222	221	218
3	0	1.82	250	248	246	242	237	235	231	231	229	225
4	-3	2.25	254	254	253	248	246	245	242	243	243	237
5	-6	2.36	258	258	258	250	248	249	246	247	247	245
6	-10	2.98	272	272	271	267	265	265	264	264	264	263

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	238	237	237	236	277	278	278	274	271	268	250	243
2	215	218	200	219	253	252	248	239	238	228	223	223
3	221	221	220	219	248	245	245	243	240	233	226	224
4	231	231	228	225	249	249	250	250	250	246	230	226
5	236	235	230	221	253	253	254	254	254	254	238	208
6	258	253	244	228	268	268	268	269	269	269	257	203

Table C-8 Deduced Experimental Results, 15.20 Percent Ethanol Concentration, 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	6.70	319	316	315	305	295	293	282	283	282	278
2	45	5.21	299	297	296	283	264	261	259	260	258	256
3	0	3.73	290	289	288	279	276	276	273	269	267	263
4	-3	4.40	296	296	296	292	288	289	286	286	283	278
5	-6	5.42	310	310	310	306	302	302	298	301	302	298
6	-10	7.05	331	331	331	326	322	322	321	322	321	318

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	274	272	275	272	311	312	313	295	286	281	277	276
2	253	254	258	258	291	291	291	269	267	262	262	261
3	260	261	261	259	283	282	282	281	280	273	263	262
4	271	271	271	268	293	293	293	293	295	288	270	272
5	291	289	286	281	308	308	309	309	309	308	293	274
6	312	311	305	294	326	326	326	326	327	327	316	265

Table C-9 Deduced Experimental Results, 21.02 Percent Ethanol Concentration, 50 and 75 Watts

Run No. 1-3 at 50 Watts, Run No. 4-5 at 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	4.40	286	285	283	268	258	248	245	248	245	243
2	0	1.95	241	238	236	230	226	223	223	223	218	214
3	-10	2.97	272	272	271	267	265	265	264	265	265	262
4	90	9.84	338	339	337	321	297	295	290	292	293	291
5	0	4.20	286	285	282	275	269	266	262	262	262	257
6	-10	8.13	340	340	340	337	331	331	330	331	330	328

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	241	243	246	246	279	279	278	267	266	258	253	252
2	212	212	212	212	238	234	234	233	230	222	218	218
3	254	247	238	212	268	268	268	268	268	268	250	249
4	290	290	290	291	334	335	334	317	315	306	302	300
5	253	253	254	254	279	277	277	275	272	264	261	261
6	320	314	306	299	335	335	336	336	336	336	318	271

Table C-10 Deduced Experimental Results, 24.06 Percent Ethanol Concentration, 50 and 75 Watts

Run No. 1-3 at 50 Watts, Run No. 4-6 at 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	3.72	274	273	271	254	239	239	236	236	236	236
2	0	2.09	238	238	235	230	222	221	218	218	218	216
3	-10	3.18	275	275	275	272	268	268	268	267	267	265
4	90	6.57	309	307	305	290	290	270	267	270	268	265
5	0	4.74	292	291	289	281	272	270	266	266	265	262
6	-10	8.50	340	340	339	335	330	330	326	326	328	325

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	236	237	238	238	269	269	268	259	254	249	247	246
2	214	214	214	214	435	234	233	231	228	203	201	201
3	253	244	237	227	270	271	271	272	272	212	249	205
4	264	266	268	268	300	301	301	300	300	312	311	311
5	258	258	258	258	286	234	283	280	273	271	265	265
6	318	312	303	294	332	222	334	330	335	335	316	263

Table C-11 Deduced Experimental Results, 40.04 Percent Ethanol Concentration, 50 and 75 Watts

Run No. 103 at 50 Watts, Run No. 4-6 at 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	3.38	247	246	244	235	225	225	224	226	226	224
2	0	2.29	227	228	226	221	215	214	213	211	212	209
3	-10	3.38	274	275	274	272	268	269	265	267	263	250
4	90	7.80	304	302	299	288	277	277	275	278	275	272
5	0	5.08	276	278	274	267	261	259	258	259	258	255
6	-10	8.55	343	344	343	339	334	336	332	333	332	323

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	223	226	227	227	241	241	241	233	233	233	233	233
2	207	210	210	210	222	223	222	223	221	218	218	217
3	233	228	226	219	273	273	274	274	275	264	237	196
4	273	270	270	270	294	295	294	283	283	283	282	282
5	253	254	254	254	270	271	270	267	266	264	263	262
6	301	300	202	286	330	338	341	341	342	332	301	247

Table C-12 Deduced Experimental Results, Pure Ethanol, 50 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	5.15	259	259	259	251	252	252	250	253	252	250
2	45	4.20	247	246	246	239	239	239	235	235	238	237
3	0	3.45	235	235	235	228	228	224	224	224	224	221
4	-3	3.72	241	240	239	233	232	232	227	227	226	225
5	-6	4.40	267	253	248	243	241	241	235	235	236	235
6	-10	2.09	332	332	302	236	200	198	198	197	197	192

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	249	252	250	250	256	256	257	258	259	260	260	260
2	236	237	237	237	241	242	243	243	243	244	244	244
3	220	220	221	219	220	220	220	220	220	220	220	220
4	225	225	224	223	234	234	235	235	234	235	235	231
5	235	230	226	224	258	250	249	248	248	248	247	204
6	186	185	179	175	322	287	241	220	212	206	202	201

Table C-13 Deduced Experimental Results, Pure Ethanol, 75 Watts

Run Number	Angle Degree	Pressure (Atm.)	External Surface Temperature of Heat Pipe									
			T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F	T_9 °F	T_{10} °F
1	90	14.00	324	324	324	315	315	316	313	314	315	312
2	45	8.50	302	303	297	292	290	286	285	283	283	283
3	0	8.06	298	298	291	277	276	276	277	275	273	273
4	-3	9.57	310	299	296	291	290	290	280	281	282	282
5	-6	7.25	381	387	330	291	271	270	261	264	264	263
6	-10	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -

Run Number	Ext. Temperature				Internal Vapor Temperature of Heat Pipe							
	T_{11} °F	T_{12} °F	T_{13} °F	T_{14} °F	T_1 °F	T_2 °F	T_3 °F	T_4 °F	T_5 °F	T_6 °F	T_7 °F	T_8 °F
1	312	312	307	306	319	320	320	322	324	324	324	324
2	278	278	278	275	294	294	294	295	295	295	295	295
3	273	273	273	273	280	280	278	278	278	278	280	278
4	277	277	277	276	296	295	295	295	295	295	295	286
5	258	257	256	255	360	314	289	282	278	274	272	222
6	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -